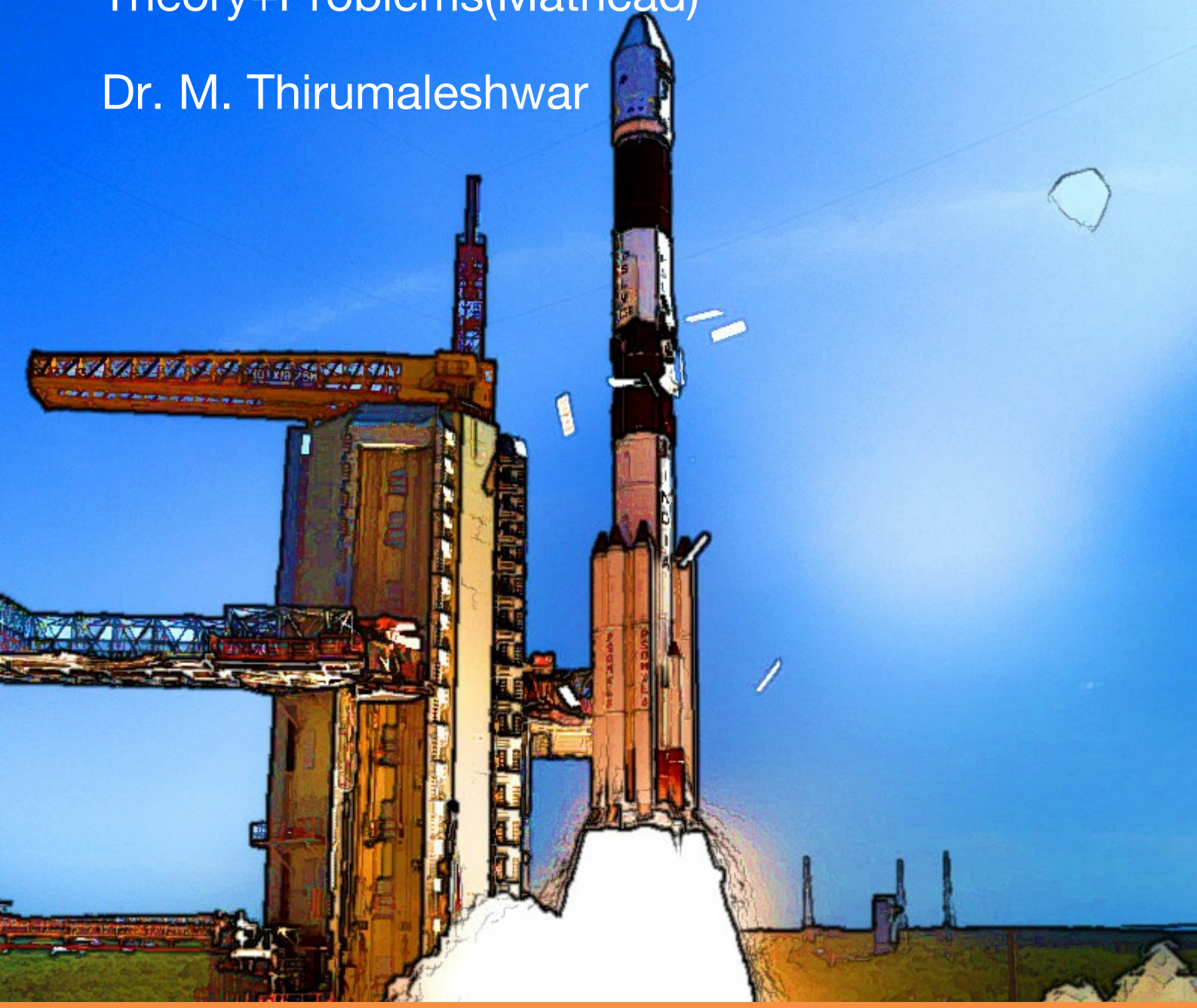
Cryogenic Engineering: Software Solutions Part-V A

Storage-Insulations-fluid transfer - Theory+Problems(Mathcad)

Dr. M. Thirumaleshwar



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CRYOGENIC ENGINEERING: SOFTWARE SOLUTIONS PART-V A

STORAGE-INSULATIONS-FLUID TRANSFER -THEORY+PROBLEMS(MATHCAD)

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Cryogenic Engineering: Software Solutions Part-V A: Storage-Insulations-fluid transfer -Theory+Problems(Mathcad)
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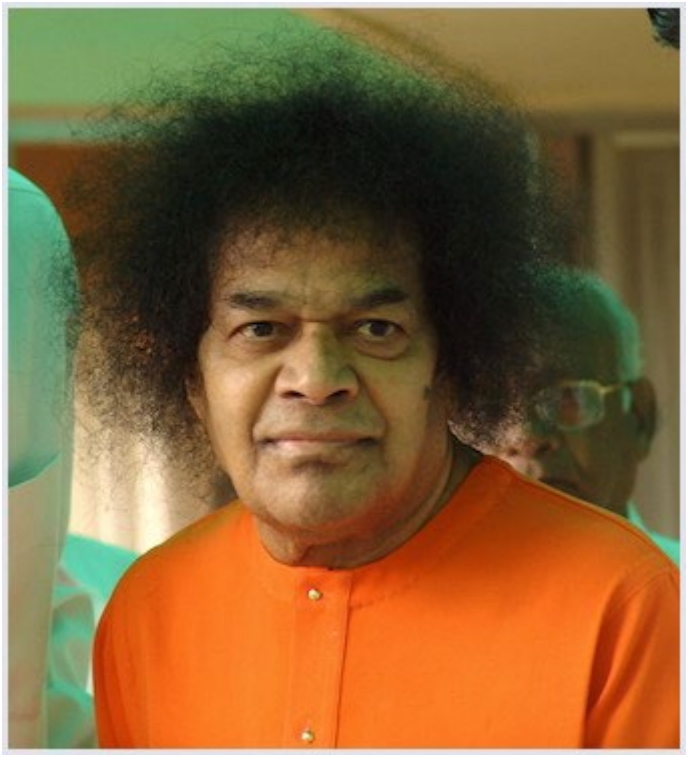
DEDICATION

This work is lovingly dedicated at the lotus feet of

Bhagavan Sri Sathya Sai Baba



"There is only one religion, the religion of Love. There is only one caste, the caste of Humanity. There is only one language, the language of the Heart. There is only one God, He is Omnipresent."

"Help Ever, Hurt Never!"

……Bhagavan Sri Sathya Sai Baba

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IN MEMORIAM....

Sweetie (13/5/2007 --- 3/10/2019)

Our pet, friend and companion, who always remained by my side during writing of all my e-books

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PREFACE

This book, viz. **Cryogenic Engineering: Software solutions – Part-V,** is the fifth in the series on **Cryogenic Engineering: Software Solutions.**

This book deals with the topics on **Cryogenic Fluid storage, insulations, and transfer systems.**

It is being published in two parts: Part- V A and Part- V B.

Part-V A contains, essential theory, data and Problems solved in Mathcad.

Part-V B contains the problems solved with Engineering Equation Solver (EES).

This part, i.e. Part-V A, contains a brief summary of background theory, definitions and formulas and the problems solved with Mathcad. As with the Part-I, II, III and IV of the series, the focus is on the solutions of problems in Cryogenic engineering using software such as Mathcad and EES. *Only the essential theory and summary of equations required for calculations are given at the beginning of the chapter*.

Advantages of using computer software to solve problems are reiterated:

1. It helps in solving the problems fast and accurately
2. Parametric analysis (what – if analysis) and graphical visualization is done very easily. This helps in an in–depth analysis of the problem.
3. Once a particular type of problem is solved, it can be used as a *template* and solving similar problems later becomes extremely easy.
4. In addition, one can plot the data, curve fit, write functions for various properties or calculations and re-use them.
5. These possibilities create interest, curiosity and wonder in the minds of students and enthuse them to know more and work more.

In this book, following topics are dealt with:

Storage vessels, Insulations, and Cryogenic Fluid transfer systems.

Under Storage vessels, we discuss: Design of inner and outer vessels, piping, draining and safety devices

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Under Insulations, we cover: Expanded foam, Gas filled powders and fibrous insulations, Vacuum insulation, Evacuated powder and fibrous insulations, Opacified powder insulation, Multilayer insulations. Also, a Table gives comparison of insulations. Important topic of vapor-shielded vessels is also covered.

Under Cryogenic Fluid transfer systems, we explain: Vacuum insulated lines, Bayonet joints for lines, cryogenic valves. Two-phase flow in transfer lines and pressure drop calculations and the cool down time and liquid requirements for transfer lines are also dealt with.

Several Functions are written in Mathcad and EES to simplify the standard and most required calculations. Students, teachers, researchers and professionals may find them very useful.

Many numerical problems, mostly from the standard Ref.[1], are solved to illustrate the ease of calculations and graphing using Mathcad and EES software.

S. I. Units are used throughout this book. Wide variety of worked examples presented in the book should be useful for those appearing for University, AMIE and Engineering Services examinations.

Acknowledgements: Firstly, I would like to **thank all my students**, who have been an inspiration to me in all my academic efforts.

Sincere thanks are due to **Rev. Fr. Joseph Lobo**, Director, St. Joseph Engineering College, Mangalore, for his kindness, regard and words of encouragement.

I am also thankful to **Dr. Thirumaleshwara Bhat**, Principal, Sri Madhwa Vadiraja Institute of Technology and Management, Bantakal, Udupi, for giving me support in my academic activities.

I gratefully remember my former colleagues at the Cryogenics section of Technical Physics Division, Bhabha Atomic Research Centre (BARC), Bombay and Centre for Advanced Technology, Indore for their sincere cooperation in a true spirit of team-work in all the projects that we undertook.

I particularly salute and admire the vision and foresight of former Heads of Technical Physics Division, BARC viz. late Mr. C. Ambasankaran, Mr. R.Y. Deshpande, Dr. S.R.Gowariker and late Mr. S.S.Ramamurthy in initiating and guiding many of the 'first of its kind' projects for Indian Space Research Organization (ISRO), wherein the Cryogenics section was deeply involved in the design and execution of the projects.

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I am especially grateful to Prof. R.G.Scurlock, former Director of Institute of Cryogenics, University of Southampton, (U. K.) for writing a message for the Part-I of this series.

It was indeed gracious of my former Professor, under whom I studied for M.Sc. in Cryogenics at the University of Southampton, U.K. during 1970 – 72, and worked as a Visiting Research Fellow during 1993-94, to honor me by writing this message.

My special thanks to **Bookboon.com** for publishing this *free ebook*. **Ms Karin Jakobsen** and the editorial staff have been most patient and helpful.

Finally, I would like to express my sincere thanks and appreciation to my **wife, Kala,** who, as usual, has given me continuous support, help and encouragement in all my academic activities, making many silent sacrifices.

M. Thirumaleshwar

September, 2019

Email: tmuliya@rediffmail.com

Note: **Along with this book, three ZIP files are also available for free download. They contain 20 'stand alone' EXE files to find properties of cryogenic fluids, design of dewar vessels, material properties, heat transfer through insulations, flow regimes and pressure drop in two phase flow, and cool down time and amount of fluid required in cooling cryogenic lines. They should be of great use for students, teachers and researchers for quick verification of their design calculations.**

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ABOUT THE AUTHOR

Dr. M. Thirumaleshwar graduated in Mechanical Engineering from Karnataka Regional Engineering College, Surathkal, Karnataka, India, in the year 1965. He obtained M.Sc (cryogenics) from University of Southampton, U.K. and Ph.D.(cryogenics) from Indian Institute of Science, Bangalore, India.

He is a Fellow of Institution of Engineers (India), Life Member, Indian Society for Technical Education, and a Foundation Fellow of Indian Cryogenics Council.

He has worked in India and abroad on large projects in the areas involving heat transfer, fluid flow, vacuum system design, cryo-pumping etc.

He worked as Head of Cryogenics Dept. in Bhabha Atomic Research Centre (BARC), Bombay and Centre for Advanced Technology (CAT), Indore, from 1966 to 1992.

He worked as Guest Collaborator with Superconducting Super Collider Laboratory of Universities Research Association, in Dallas, USA from 1990 to 1993.

He also worked at the Institute of Cryogenics, Southampton, U.K. as a Visiting Research Fellow from 1993 to 1994.

He was Head of the Dept. of Mechanical Engineering, Fr. Conceicao Rodrigues Institute of Technology, Vashi, Navi Mumbai, India for eight years.

He also worked as Head of Dept. of Mechanical Engineering and Civil Engineering, and then as Principal, Vivekananda College of Engineering and Technology, Puttur (D.K.), India.

He was Professor and coordinator of Post-graduate program in the Dept. of Mechanical Engineering in St. Joseph Engineering College, Vamanjoor, Mangalore, India.

A book entitled **"Fundamentals of Heat and Mass Transfer"** authored by him and published by M/s Pearson Education, India (2006) **has been adopted as a Text book** for third year engineering students by the Visweswaraya Technological University (V.T.U.), Belgaum, India.

He has authored a ***free e-book*** entitled **"Software Solutions to Problems on Heat Transfer"** wherein problems are solved using 4 software viz. Mathcad, EES, FEHT and EXCEL. This book, containing about 2750 pages, is presented in 9 parts and all the 9 parts can be downloaded *for free* from www.bookboon.com

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He has also authored ***free e-books*** **on Thermodynamics** entitled **"Basic Thermodynamics: Software Solutions" and "Applied Thermodynamics: Software Solutions"** wherein problems are solved using 3 software viz. Mathcad, EES, and TEST. Each of these titles is presented in 5 parts and all the books can be downloaded *for free* from www.bookboon.com

In addition, he has authored following two useful *free ebooks*:

1. Applied Thermodynamics: Software Solutions: Vapor compression Refrigeration cycle + Problems (Mathcad)
   http://bookboon.com/en/applied-thermodynamics-software-solutions-vapor-ebook
2. Applied Thermodynamics: Software Solutions: Vapor Power cycles (Rankine cycle) + Problems (Mathcad)
   http://bookboon.com/en/applied-thermodynamics-software-solutions-vapor-po-ebook

His earlier *free ebooks in this* **Cryogenic Engineering series**, viz. **Cryogenic Engineering: Software Solutions – Parts-I, II-A and II-B, III-A, III-B and IV** were published by Bookboon recently.

He has also authored **three motivational, free ebooks**, published by www.bookboon.com, entitled as follows:

1. Towards Excellence… How to Study (A Guide book to Students)
2. Towards Excellence… How to teach (A guide book to Teachers)
3. Towards Excellence… Seminars, GD's and Personal Interviews

Dr. M. Thirumaleshwar has attended several National and International conferences and has more than 50 publications to his credit.

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ABOUT THE SOFTWARE USED

Following software are used while solving problems in this book series:

1. Mathcad 7 and Mathcad 15 (Ref: www.ptc.com)
2. Engineering Equation Solver (EES) (Ref: www.fchart.com), and

For a brief introduction to Mathcad, EES and EXCEL see the chapter 1 of the following *free ebook* by the author:

"Software Solutions to Problems on Heat Transfer – CONDUCTION-Part-I":
http://bookboon.com/en/software-solutions-to-problems-on-heat-transfer-ebook

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6 CRYOGENIC FLUID STORAGE, INSULATIONS, AND TRANSFER SYSTEMS

Learning objectives:

1. In this chapter, topics on **'Cryogenic Fluid Storage, Insulations, and Transfer systems'** are dealt with.
2. First, essential parts of a dewar vessel are are explained.
3. Next, design of inner and outer vessels are taken up. And, the necessary data tables for design are presented.
4. Several Mathcad Functions are written to quickly determine the required material properties. Mathcad Functions are also written for the design of inner and outer vessels, cylindrical/spherical heads etc. Problems are solved to illustrate the use of these Mathcad Functions.
5. Then, we explain: draining of dewar vessels and pressurization mass requirements, with Mathcad Functions and solved problems.
   And, the various properties of cryogenic fluids required to solve problems in this book are taken from the NIST website, i.e. https://webbook.nist.gov/chemistry/fluid/
6. Various types of insulations used in Cryogenics are discussed next; many Mathcad Functions are written, and problems solved.
7. Vapor-shielded vessels, vacuum insulated transfer lines, bends, thermal strain in lines, bayonet joints, valves etc are explained next. Very useful Mathcad Functions are written for vapor shielded vessels, thermal contraction etc, and many problems solved.
8. Two-phase flow, various flow regimes, and pressure drop in two-phase flow are explained next. Problems are solved to illustrate the Lockhart-Martinelli correlation to determine pressure drop.
9. Next, cool down time for cryogenic lines, amount of cryogenic liquid required to cool down are dealt with, and Mathcad Functions written, and problems solved.
10. Several Functions/ Procedures are written in Mathcad/EES which greatly simplify the calculations. These should be very much useful to teachers, students and researchers working in this field.

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6.1 CRYOGENIC FLUID STORAGE VESSELS[1-12]

Introduction [1]:

Sizes of Cryogenic fluid storage vessels range from 1-liter flasks used in laboratory work up to 106 m^3 (28,000 gal U.S.). Larger vessels are used to store liquid nitrogen, liquid oxygen, and liquid hydrogen for industrial use and in space-vehicle ground support systems. Storage vessels are insulated by rigid foam or fibrous insulation, or with multilayer evacuated insulations (so that less than 0.1 percent of the vessel contents is lost per day).

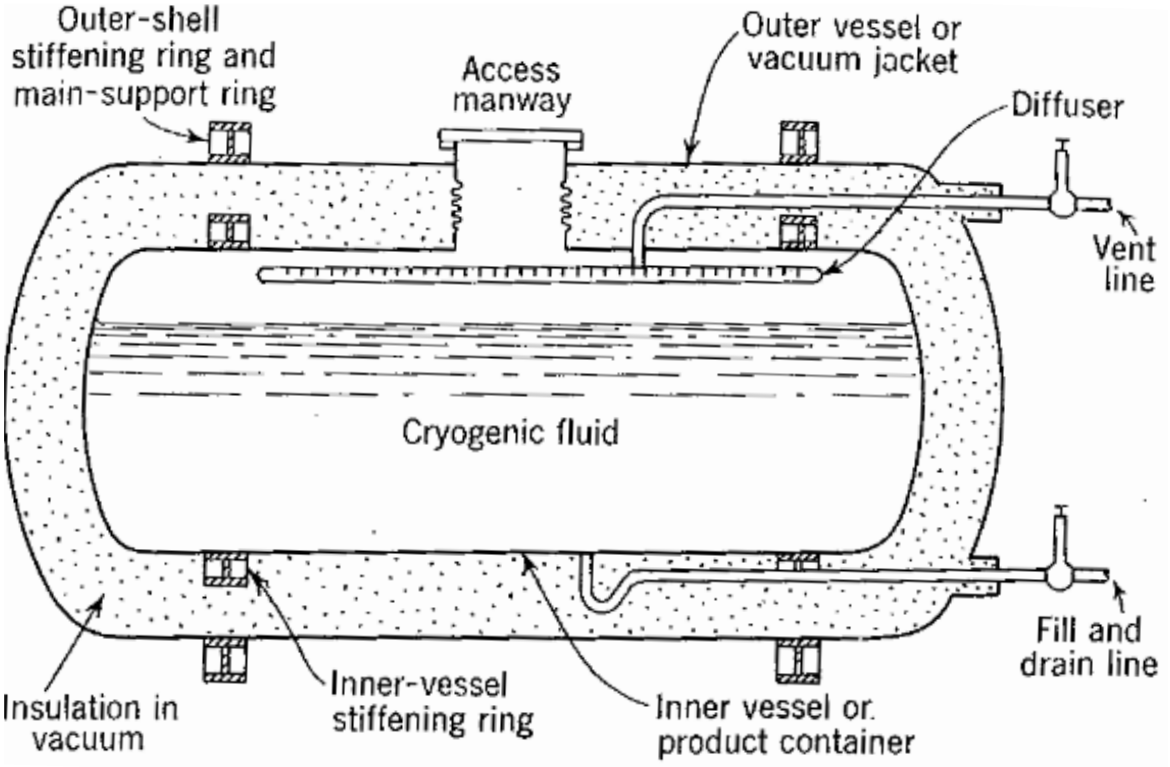

Fig. 6.1. Elements of a dewar vessel.

The essential elements of a dewar vessel are shown in Fig. 5.1. The storage vessel consists of an inner vessel which contains the cryogenic fluid to be stored. The inner vessel is enclosed by an outer vessel or vacuum jacket, which contains the foam, or high vacuum, or multilayer insulation as the case may be.

In small laboratory dewars, the “insulation” consists of the silvered walls and high vacuum alone. Insulations such as powders, fibrous materials, or multilayer insulations ‘are used in larger vessels.

A fill and drain line (which may be two separate lines or a single line) is necessary for larger vessels. A vapor vent line must be provided to allow the vapor formed as a result of heat inleak to escape. In addition, some method must be provided to remove the liquid from

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the container. Liquid removal may be accomplished either by pressurization of the inner container or by a liquid pump. Anti-slosh baffles are employed in transportable vessels to damp motion of the liquid while the container is being moved. A suspension system must be used to support the product container within the vacuum jacket.

Cryogenic-fluid storage vessels are not designed to be completely filled to prevent rapid pressure rise due to evaporation of liquid. A 10 percent ***ullage volume*** is commonly used for large storage vessels. This means that a nominal 106 m^3 dewar actually has an internal volume of 116.6 m^3.

Cryogenic-fluid storage vessels may be constructed in almost any shape: cylindrical, spherical, conical, or any combination of these shapes. Generally, one of the most economical configurations is the cylindrical vessel with either dished, elliptical, or hemispherical heads or end closures. Spherical vessels have the most effective configuration as far as heat inleak is concerned, and they are often used for large-volume storage in which the vessel is constructed on the site.

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Cylindrical vessels are usually required for transportable trailers and railway cars because the outside diameter of the vessel cannot exceed about 2.44 m for normal highway transportation. For shop-fabricated stationary vessels that are shipped to the site by rail, the maximum diameter depends upon the route taken, but generally not more than 3.05 m to 3.66 m can be accommodated.

Some typical examples of Dewar vessels:

1. **Portable dewars for Liquid Nitrogen, Oxygen and Argon: capacity: 15 L to 500 L. from M/s Cryofab.** (https://www.cryofab.com/products/cl_clpb_series)

2. **Portable Liquid helium dewars from Cryofab: capacities: 30, 60, 100, 150, 200, 250, 350, 500 and 1000 Litres.**
   Ref: https://www.cryofab.com/products/cmsh_series

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3. **Transport trailers from Air Liquide:**
   Ref: https://www.airliquide.com/industry/supply-modes

6.2 DESIGN OF INNER-VESSEL AND OUTER VESSELS

Inner vessel:

The details of conventional cryogenic-fluid storage- vessel design are covered in such standards as the ASME Boiler and Pressure Vessel Code, Section VIII (1983), and the British Standards Institution Standard 1500 or 1515. Most users require that the vessels be designed, fabricated, and tested according to the Code for sizes larger than about 250 dm^3 (66 gal U.S.) because of the proved safety of Code design.

The inner vessel must withstand the design internal pressure, the weight of the fluid within the vessel, and the bending stresses as a result of beam-bending action. The inner vessel must be constructed of a material compatible with the cryogenic fluid; therefore, stainless steel, aluminum, Monel, and in some cases copper are commonly used for the inner shell. These materials are much more expensive than ordinary carbon steel, so the designer would like to make the inner-vessel wall as thin as practical in order to hold the cost within reason. In addition, a thick walled vessel requires a longer time to cool down, wastes more liquid in cool-down, and introduces the possibility of thermal stresses in the vessel wall during cool-down. For these reasons, the inner vessel is designed to withstand only the internal pressure and bending forces, and stiffening rings are used to support the weight of the fluid within the lower vessel.

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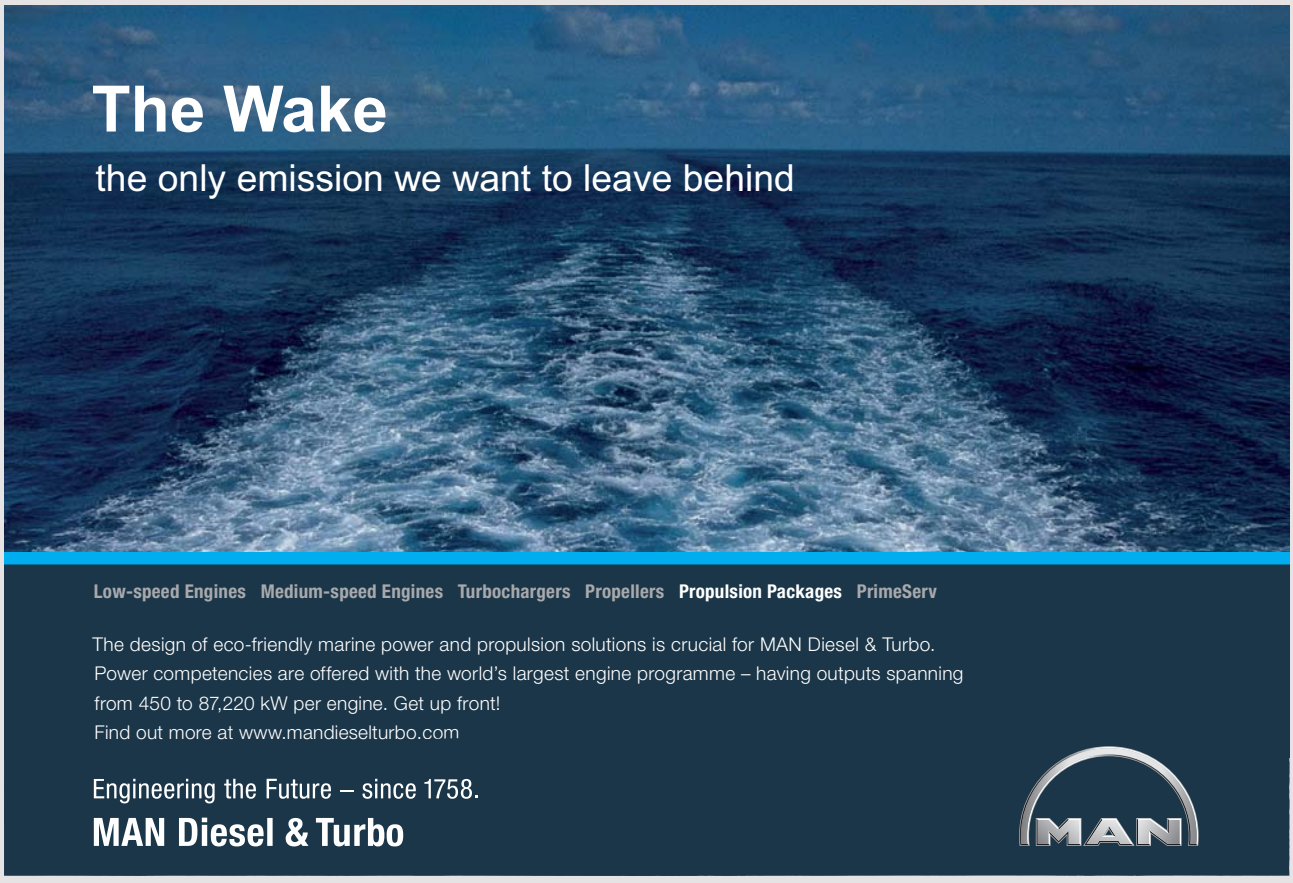
Outer-vessel:

Because the outer shell of a dewar vessel has atmospheric pressure acting on it. Actually, the outer shell would not fail because of excessive stress, but it would fail by buckling. (A thin cylindrical shell will collapse under an externally applied pressure at stress values much lower than the yield strength for the material). So, outer vessel should be designed accordingly.

Note: Complete design of a dewar vessel requires the design of inner vessel, outer vessel, support system stiffening rings for inner and outer vessels, consideration of acceleration forces during transportation, etc. **However, here we will deal with the design of inner and outer vessels only; for other aspects of design, the reader may refer to the standard text book by Barron [1].**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6.2.1 SOME ESSENTIAL DATA FOR DESIGN [1]

Following Tables give some essential data:

Table 6.1 Allowable stress for materials at room temperature or lower (ASME Code, Section VIII, 1983)

Material		Allowable Stress MPa	Allowable Stress psi
Carbon steel (for outer shell only)	SA-285 Grade C	94.8	13 750
	SA-299	129.2	18 750
	SA-442 Grade 55	94.8	13 750
	SA-516 Grade 60	103.4	15 000
Low-alloy steel	SA-202 Grade B	146.5	21 250
	SA-353-B (9% Ni)	163.7	23 750
	SA-203 Grade E	120.6	17 500
	SA-410	103.4	15 000
Stainless steel	SA-240 (304)	129.2	18 750
	SA-240 (304L)	120.6	17 500
	SA-240 (316)	129.2	18 750
	SA-240 (410)	112.0	16 250
Aluminum	SB-209 (1100-0)	16.2	2 350
	SB-209 (3004-0)	37.9	5 500
	SB-209 (5083-0)	68.9	10 000
	SB-209 (6061-T4)	41.4	6 000
Copper	SB-11	46.2	6 700
	SB-169 (annealed)	86.2	12 500
Nickel alloys (annealed)	SB-127 (Monel)	128.2	18 600
	SB-168	137.9	20 000

Table 6.2 Weld efficiencies for arc-welded and gas-welded joints (ASME Code, Section VIII, 1983)

Type of Joint	Fully Radiographed	Spot Examined	Not Spot Examined
Butt joints with complete penetration	1.00	0.85	0.70
Single-welded butt joint, backing strip	0.90	0.80	0.65
Single-welded butt joint, no backing strip	...	...	0.60
Double full-fillet lap joint	...	...	0.55

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Table 6.3 Geometric characteristics of heads: D = inside diameter, D_o = outside diameter, t_h = thickness, R = dish radius (ASME torispherical)

Hemispherical	2:1 Elliptical	ASME Torispherical
Internal volume $V = \pi D^3/12$	$V = \pi D^3/24$	$V = \pi R^3/39.173$
Material volume $V_h = \frac{1}{2}\pi(D + t_h)^2 t_h$	$V_h = 0.3450\pi(D + t_h)^2 t_h$	$V_h = 0.264\pi(R + t_h)^2 t_h$
Outside surface area $A_0 = \frac{1}{2}\pi D_o^2$	$A_0 = 0.3450\pi D_o^2$	$A_0 = 0.264\pi R^2$

The dish radius for the ASME torispherical head is related to the outside diameter of the head as follows: (1) for $D_0 \leqq 42$ in. (1.069 m), $R = D_0 - 2t_h$; (2) for 42 in. $< D_0 <$ 144 in. (3.658 m), $R = D_0 - 2t_h - 6$ in.; (3) for $D_0 \geqq 144$ in., $R = D_0 - 2t_h - 12$ in.

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Table 6.4 Standard thicknesses for sheet, plate, and heads. Thickness for metric elements is in mm; thickness for conventional elements is in inches.

Sheet		Plate		Heads	
Metric	Conven.	Metric	Conven.	Metric	Conven.
2.4	13 ga. (0.094)	5	3/16	5	3/16
2.6	...	6	1/4	6	1/4
2.8	12 ga. (0.109)	7	...	8	5/16
3.0	...	8	5/16	10	3/8
3.2	11 ga. (0.125)	9	...	11	7/16
3.4	...	10	3/8	12	1/2
3.6	10 ga. (0.141)	11	7/16	14	9/16
3.8	...	12	1/2	16	5/8
4.0	...	14	9/16	18	...
4.2	...	16	5/8	20	3/4
4.5	8 ga. (0.172)	18	...	22	7/8
4.8	...	20	3/4	25	1
5.0	...	22	7/8	28	1 1/8
		25	1	30	...
		28	1 1/8	32	1 1/4
		30	...	35	1 3/8
		32	1 1/4	38	1 1/2
		35	1 3/8	40	1 5/8
		38	1 1/2	45	1 3/4
		40	1 5/8	50	2
		45	1 3/4		
		50	2		

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Table 6.5 Typical mechanical properties of metals

Metal	Density		Young's Modulus		Poisson's Ratio
	kg/m^3	lb_m/in^3	GPa	psi	
Carbon steel	7720	0.279	200	29×10^6	0.27
Low-alloy steel	7830	0.283	200	29×10^6	0.27
Stainless steel	7920	0.286	207	30×10^6	0.28
Aluminum	2700	0.098	69	10×10^6	0.33
Copper	8940	0.323	117	17×10^6	0.33
Monel	8830	0.319	179	26×10^6	0.32

Table 6.6 Equivalent radius for elliptical heads under external pressure. $R_0 = K_1 D$ and D = major diameter of the ellipse, D_1 = minor diameter of the ellipse.

D/D_1	K_1	D/D_1	K_1
3.0	1.36	1.8	0.81
2.8	1.27	1.6	0.73
2.6	1.18	1.4	0.65
2.4	1.08	1.2	0.57
2.2	0.99	1.0	0.50
2.0	0.90		

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Table 6.7 Thermal Conductivities and Thermal Conductivity Integrals for Several Materials[a]

	Cu (beryllium tough pitch)		Aluminium (6063 T5)		Low-carbon steel (C1020)		Stainless steel (304)		Monel (drawn)		Teflon	
Temperature (K)	K_t	$\int_{4.2\,K}^{T} K_t\,dT$	K_t	$\int_{4.2\,K}^{T} K_t\,dT$	K_t	$\int_{4.2\,K}^{T} K_t\,dT$	K_t	$\int_{4.2\,K}^{T} K_t\,dT$	K_t	$\int_{4.2\,K}^{T} K_t\,dT$	K_t	$\int_{4.2\,K}^{T} K_t\,dT$
4.2	1.9	0	34	0	3.0	0	0.24	0	0.43	0	0.046	0.0
10	4.8	19	86	360	11.5	43	0.77	2.9	1.74	6.3	0.096	0.44
20	10.6	95	170	1650	24.0	222	1.95	16.3	4.30	36.4	0.141	1.64
30	16.2	229	230	3650	32.0	502	3.30	42.4	6.90	92.9	0.174	3.23
40	21.0	415	270	6200	38.6	867	4.70	82.4	9.00	173	0.193	5.08
50	26.1	650	280	8950	47.6	1310	5.80	135	10.95	273	0.208	7.16
60	30.0	930	270	11700	53.6	1810	6.80	198	12.09	368	0.219	9.36
70	33.7	1250	248	14300	57.5	2360	7.60	270	13.06	513	0.228	11.6
80	37.0	1600	230	16700	60.0	2950	8.26	349	13.90	647	0.235	13.9
90	40.1	1990	222	19000	61.8	3350	8.86	436	14.63	791	0.241	16.3
100	43.0	2400	216	21100	62.9	4170	9.40	528	15.27	940	0.245	18.7
120	48.4	3300	207	25300	64.1	5450	10.36	726	16.26	1260	0.251	23.7
140	53.3	4320	201	29300	64.6	6750	11.17	939	17.34	1590	0.255	28.7
160	57.6	5440	200	33300	64.8	8050	11.86	1170	18.25	1950	0.257	33.8
180	61.5	6640	200	37300	64.9	9350	12.47	1410	19.02	2320	0.258	39.0
200	65.0	7910	200	41300	65.0	10700	13.00	1660	19.69	2710	0.259	44.2
250	72.4	11300	200	51300	65.0	13900	14.07	2340	21.02	3730	0.260	57.2
300	78.5	15000	200	61300	65.0	17200	14.90	3060	22.00	4800	0.260	70.2

[a] The thermal conductivity (K_t) is expressed in dimensions of W/(m K), and the thermal conductivity integral ($\int_{4.2\,K}^{T} K_t\,dT$) is given in dimensions of W/m.

Note: Above Table is from Ref.[2]

Table 6.8 Stainless-steel IPS pipe sizes

Nominal Size (in.)	OD (in.)	Wall Thickness (in.)			
		Schedule 5	Schedule 10	Schedule 40	Schedule 80
½	0.840	0.065	0.083	0.109	0.147
¾	1.050	0.065	0.083	0.113	0.154
1	1.315	0.065	0.109	0.133	0.179
1¼	1.660	0.065	0.109	0.140	0.191
1½	1.900	0.065	0.109	0.145	0.200
2	2.375	0.065	0.109	0.154	0.218
2½	2.875	0.083	0.120	0.203	0.276
3	3.500	0.083	0.120	0.216	0.300
3½	4.000	0.083	0.120	0.226	0.318
4	4.500	0.083	0.120	0.237	0.337
5	5.563	0.109	0.134	0.258	0.375
6	6.625	0.109	0.134	0.280	0.432
8	8.625	0.109	0.148	0.322	0.500
10	10.750	0.134	0.165	0.365	0.500
12	12.750	0.156	0.180	0.375	0.500
14	14.000	0.156	0.188	0.375	0.500
16	16.000	0.165	0.188	0.375	0.500
18	18.000	0.165	0.188	0.375	0.500
20	20.000	0.188	0.218	0.375	0.500
22	22.000	0.188	0.218	0.375	0.500
24	24.000	0.218	0.250	0.375	0.500

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Table 6.9 Stainless-steel IPS pipe sizes (metric)

Nominal Size (mm)	OD (mm)	Wall Thickness (mm)			
		Schedule 5	Schedule 10	Schedule 40	Schedule 80
15	21.3	1.7	2.1	2.8	3.7
20	26.7	1.7	2.1	2.9	3.9
25	33.4	1.7	2.8	3.4	4.6
32	42.2	1.7	2.8	3.6	4.9
40	48.3	1.7	2.8	3.7	5.1
50	60.3	1.7	2.8	3.9	5.5
65	73.0	2.1	3.1	5.2	7.0
80	101.6	2.1	3.1	5.7	8.1
100	114.3	2.1	3.1	6.0	8.6
125	141.3	2.8	3.4	6.6	9.5
150	168.3	2.8	3.4	7.1	11.0
200	219.1	2.8	3.8	8.2	12.7
250	273.1	3.4	4.2	9.3	12.7
300	323.9	4.0	4.6	9.5	12.7
350	355.6	4.0	4.8	9.5	12.7
400	406.4	4.2	4.8	9.5	12.7
450	457.2	4.2	4.8	9.5	12.7
500	508.0	4.8	5.5	9.5	12.7
550	558.8	4.8	5.5	9.5	12.7
600	609.6	5.5	6.4	9.5	12.7

Another useful extract from a chart to determine the wall thickness of a pipe is given below:

Ref: https://www.nandishwarsteel.com/pdf/pipes-new.pdf

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STAINLESS STEEL PIPE DIMENSION AS PER ASTM & WEIGHT-KG. PER MTR. (ANSI B36.19)

Nominal Bore		Outside Diameter	Sch-5S		Sch-10S		Sch-40S		Sch-80S		Sch-160S		Sch-XXS	
mm	INCH	mm	Wt mm	Weight (Kg/mt)	Wt mm	Weight (Kg/mt)	Wt mm	Weight (Kg/mt)	Wt mm	Weight (Kg/mt)	Wt mm	Weight (Kg/mt)	Wt mm	Weight (Kg/mt)
3	1/8	10.3	1.24	0.276	1.24	0.28	1.73	0.37	2.41	0.47	-	-	-	-
6	1/4	13.7	1.24	0.390	1.65	0.49	2.24	0.631	3.02	0.80	-	-	-	-
10	3/8	17.1	1.24	0.490	1.65	0.63	2.31	0.845	3.20	1.10	-	-	-	-
15	1/2	21.3	1.65	0.800	2.11	1.00	2.77	1.27	3.75	1.62	4.75	1.94	7.47	2.55
20	3/4	26.7	1.65	1.03	2.11	1.28	2.87	1.68	3.91	2.20	5.54	2.89	7.82	3.63
25	1	33.4	1.65	1.30	2.77	2.09	3.38	2.50	4.55	3.24	6.35	4.24	9.09	5.45
32	1 1/4	42.2	1.65	1.65	2.77	2.70	3.56	3.38	4.85	4.47	6.35	5.61	9.70	7.77
40	1 1/2	48.3	1.65	1.91	2.77	3.11	3.68	4.05	5.08	5.41	7.14	7.25	10.16	9.54
50	2	60.3	1.65	2.40	2.77	3.93	3.91	5.44	5.54	7.48	8.74	11.1	11.07	13.44
65	2 1/2	73.0	2.11	3.69	3.05	5.26	5.16	8.63	7.01	11.4	9.53	14.9	14.2	20.39
80	3	88.9	2.11	4.51	3.05	6.45	5.49	11.30	7.62	15.2	11.1	21.3	15.24	27.65
100	4	114.3	2.11	5.84	3.05	8.36	6.02	16.07	8.56	22.3	13.49	33.54	17.12	41.03
125	5	141.3	2.77	9.47	3.40	11.57	6.55	21.8	9.53	31.97	15.88	49.11	19.05	57.43
150	6	168.3	2.77	11.32	3.40	13.84	7.11	28.3	10.97	42.7	18.2	67.56	21.95	79.22
200	8	219.1	2.77	14.79	3.76	19.96	8.18	42.6	12.7	64.6	23.0	111.2	22.23	107.8
250	10	273.1	3.40	22.63	4.19	27.78	9.27	60.5	12.7	96.0	28.6	172.4	25.40	155.15
300	12	323.9	3.96	31.25	4.57	36.00	9.52	73.88	12.7	132.0	33.32	238.76	25.40	186.97
350	14	355.6	3.96	34.36	4.78	41.3	11.13	94.59	19.05	158.08	35.71	281.70	-	-
400	16	406.4	4.19	41.56	4.78	47.29	12.7	123.30	21.41	203.33	40.46	365.11	-	-
450	18	457.2	4.19	46.80	4.78	53.42	14.27	155.80	23.8	254.36	45.71	466.40	-	-
500	20	508.0	4.78	59.25	5.54	68.71	15.09	183.42	26.19	311.2	49.99	564.68	-	-
600	24	609.6	5.54	82.47	6.35	94.45	17.48	255.41	30.96	442.08	59.54	808.22	-	-

6.2.2 SOME MATHCAD FUNCTIONS FOR MATERIAL PROPERTIES

We give below some Mathcad Functions to determine a few material properties with reference to the data Tables given above:

Prob. 6.1 Write a Mathcad program to find Allowable stress (Sa, MPa) for various materials used in Cryogenics applications.

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Put the data from Table 6.1 in a 3 column matrix, as shown below:

	Material	MaterialNumber	Allowable stress (MPa)
M1 :=	"Carbon steel SA-285 Grade C"	0	94.8
	"Carbon steel SA-299"	1	129.2
	"Carbon steel SA-442 Grade 55"	2	94.8
	"Carbon steel SA-516 Grade 60"	3	103.4
	"Low alloy steel SA-202 Grade B"	4	146.5
	"Low alloy steel SA-353 B (90% Ni"	5	163.7
	"Low alloy steel SA-203 Grade B"	6	120.6
	"Low alloy steel SA-410"	7	103.4
	"Stainless steel SA-240 (304)"	8	129.2
	"Stainless steel SA-240 (304L)"	9	120.6
	"Stainless steel SA-240 (316)"	10	129.2
	"Stainless steel SA-240 (410)"	11	112
	"Aluminium SB-209 (1100-0)"	12	16.2
	"Aluminium SB-209 (3004-0)"	13	37.9
	"Aluminium SB-209 (5083-0)"	14	68.9
	"Aluminium SB-209 (6061-T4)"	15	41.4
	"Copper SB-11"	16	46.2
	"Copper SB-169 (annealed)"	17	86.2
	"Nickel alloys (annealed) SB-127 (monel)"	18	128.2
	"Nickel alloys (annealed) SB-168"	19	137.9

Now, the Mathcad Function is:

Program:

Allowable_stress(MaterialNumber) := for $i \in 0..19$

$$\begin{bmatrix} \text{"MaterialNumber"} & \text{"Material"} & \text{"Allowable stress, S\_a (MPa)"} \\ M1_{i,1} & M1_{i,0} & M1_{i,2} \end{bmatrix} \text{ if MaterialNumber=i}$$

Ex:

MaterialNumber := 8

$$\text{Allowable\_stress(MaterialNumber)} = \begin{bmatrix} \text{"MaterialNumber"} & \text{"Material"} & \text{"Allowable stress, S\_a (MPa)"} \\ 8 & \text{"Stainless steel SA-240 (304)"} & 129.2 \end{bmatrix}$$

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Prob. 6.2 Write a Mathcad program to determine density and Young's Modulus for a few materials.

Put the data in Table 6.5 in a Matrix as shown below:

Material No. Material rho (kg/m^3) E_y (GPa) Poissons ratio

$$M2 := \begin{bmatrix} 0 & \text{"Carbon steel"} & 7720 & 200 & 0.27 \\ 1 & \text{"Low alloy steel"} & 7830 & 200 & 0.27 \\ 2 & \text{"Stainless steel"} & 7920 & 207 & 0.28 \\ 3 & \text{"Aluminium"} & 2700 & 69 & 0.33 \\ 4 & \text{"Copper"} & 8940 & 117 & 0.33 \\ 5 & \text{"Monel"} & 8830 & 179 & 0.32 \end{bmatrix}$$

Now, write the desired program:

Program:

Youngs_Modulus_etc(MaterialNumber) := for i ∈ 0..5

$$\begin{bmatrix} \text{"MaterialNumber"} & \text{"Material"} & \text{"Density, rho (kg/m^3)"} & \text{"Young's Mod., E_y (GPa)"} & \text{"Poisson's ratio, nu"} \\ M2_{i,0} & M2_{i,1} & M2_{i,2} & M2_{i,3} & M2_{i,4} \end{bmatrix} \text{ if MaterialNumber=i}$$

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Ex:

MaterialNumber := 2

Youngs_Modulus_etc(MaterialNumber) =

"MaterialNumber"	"Material"	"Density, rho (kg/m^3)"	"Young's Mod., E_y (GPa)"	"Poisson's ratio, nu"
2	"Stainless steel"	$7.92 \cdot 10^3$	207	0.28

==========

Prob.6.3 Write Mathcad Functions for Geometrical characteristics of Heads:

D = inside dia, D_0 = outside dia, th = thickness, R = dish radius (ASME torisph.)

Take formulas from Table 6.3:

Mathcad Programs:

$$\text{Hemisph\_head}(D, D_0, t_h) := \begin{bmatrix} V \leftarrow \frac{\pi \cdot D^3}{12} \\ V_h \leftarrow \frac{1}{2} \cdot \pi \cdot (D + t_h)^2 \cdot t_h \\ A_0 \leftarrow \frac{1}{2} \cdot \pi \cdot D_0^{\,2} \\ \begin{bmatrix} \text{"Type of Head"} & \text{"Internal Vol. (m^3)"} & \text{"Material Vol. (m^3)"} & \text{"Outside surface area (m^2)"} \\ \text{"Hemispherical"} & V & V_h & A_0 \end{bmatrix} \end{bmatrix}$$

Ex:

$D := 3.05$ m $\quad D_0 := 3.07$ m $\quad t_h := 0.01$ m

Hemisph_head(D, D_0, t_h) =

"Type of Head"	"Internal Vol. (m^3)"	"Material Vol. (m^3)"	"Outside surface area (m^2)"
"Hemispherical"	7.428	0.147	14.805

$$\text{Elliptical\_head}(D, D_0, t_h) := \begin{bmatrix} V \leftarrow \frac{\pi \cdot D^3}{24} \\ V_h \leftarrow 0.345 \cdot \pi \cdot (D + t_h)^2 \cdot t_h \\ A_0 \leftarrow 0.345 \cdot \pi \cdot D_0^{\,2} \\ \begin{bmatrix} \text{"Type of Head"} & \text{"Internal Vol. (m^3)"} & \text{"Material Vol. (m^3)"} & \text{"Outside surface area (m^2)"} \\ \text{"Elliptical"} & V & V_h & A_0 \end{bmatrix} \end{bmatrix}$$

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Ex:

$D := 3.05 \quad m \qquad D_0 := 3.07 \quad m \qquad t_h := 0.01 \quad m$

$$\text{Elliptical\_head}(D, D_0, t_h) = \begin{bmatrix} \text{"Type of Head"} & \text{"Internal Vol. (m^3)"} & \text{"Material Vol. (m^3)"} & \text{"Outside surface area (m^2)"} \\ \text{"Elliptical"} & 3.714 & 0.101 & 10.215 \end{bmatrix}$$

$$\text{ASME\_Torisph\_head}(R, t_h) := \left| \begin{array}{l} V \leftarrow \dfrac{\pi \cdot R^3}{39.173} \\ V_h \leftarrow 0.264 \cdot \pi \cdot (R + t_h)^2 \cdot t_h \\ A_0 \leftarrow 0.264 \cdot \pi \cdot R^2 \\ \begin{bmatrix} \text{"Type of Head"} & \text{"Internal Vol. (m^3)"} & \text{"Material Vol. (m^3)"} & \text{"Outside surface area (m^2)"} \\ \text{"ASME\_Torisph."} & V & V_h & A_0 \end{bmatrix} \end{array} \right.$$

Ex:

$R := 1.502 \quad m \qquad t_h := 0.01 \quad m$

$$\text{ASME\_Torisph\_head}(R, t_h) = \begin{bmatrix} \text{"Type of Head"} & \text{"Internal Vol. (m^3)"} & \text{"Material Vol. (m^3)"} & \text{"Outside surface area (m^2)"} \\ \text{"ASME\_Torisph."} & 0.272 & 0.019 & 1.871 \end{bmatrix}$$

6.2.3 DESIGN EQUATIONS FOR THICKNESS OF INNER SHELL OF A CYLINDRICAL VESSEL [1]

$$t_{shell} = \frac{p \cdot D}{2 \cdot S_a \cdot e_w - 1.2 \cdot p} = \frac{p \cdot D_0}{2 \cdot S_a \cdot e_w + 0.8 \cdot p}$$

where:

p = design internal pressure (absolute pressure for vacuum jacketed vessels)

D = inside diameter of shell

D_0 = outside diameter of shell

S_a = allowable stress (approx. one-fourth of min. ultimate strength)

e_w = weld efficiency

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6.2.4 DESIGN EQUATIONS FOR MIN. THICKNESS OF SPHERICAL SHELLS, HEMISPHERICAL HEADS, ELLIPTICAL HEADS, OR ASME TORISPHERICAL HEADS [1]

$$t_h = \frac{p \cdot D \cdot K}{2 \cdot S_a \cdot e_w - 0.2 \cdot p} = \frac{p \cdot D_0 \cdot K}{2 \cdot S_a \cdot e_w + 2 \cdot p \cdot (K - 0.1)}$$

where:

D = inside dia of spherical vessel or hemispherical head, of the inside major dia of an elliptical head, or (2 * crown radius) for ASME torispherical head.

Value of constant K is given by:

$$K = \frac{1}{6} \cdot \left[2 + \left(\frac{D}{D_1} \right)^2 \right]$$

where D_1 is the minor dia of the elliptical head.

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For ASME torispherical head, K = 0.885.

Prob.6.4 Write a Mathcad Function for min. thickness of inner shell for a cylindrical vessel surrounded by a vacuum jacket.

We have, from Ref.[1]:

$$t_{shell} = \frac{p \cdot D}{2 \cdot S_a \cdot e_w - 1.2 \cdot p} = \frac{p \cdot D_0}{2 \cdot S_a \cdot e_w + 0.8 \cdot p} \quad \text{...mm}$$

where:

p = design internal pressure (abs. pressure for vac. jacketed vessels), kPa

D = inside dia of shell, mm

D_0 = OD of shell, mm

S_a = allowable stress, kPa

e_w = weld effcy.

The Mathcad Function (based on inside dia of shell):

$$t_{inner\_shell}(p, D, S_a, e_w) := \frac{p \cdot D}{2 \cdot S_a \cdot e_w - 1.2 \cdot p} \quad \text{... mm}$$

Ex: $p := 800$ kPa $D := 3050$ mm $S_a := 129200$ kPa $e_w := 1$

Then, we get:

$t_{inner\_shell}(p, D, S_a, e_w) = 9.478$ mm .... thickness of cyl. inner shell of vac. jacketed tank

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Prob.6.5 Write a Mathcad Function for min. thickness for spherical shells, hemisph. heads, Elliptical heads, or ASME torispherical heads:

We have, from Ref. [1],

$$t_h = \frac{p \cdot D \cdot K}{2 \cdot S_a \cdot e_w - 0.2 \cdot p} = \frac{p \cdot D_0 \cdot K}{2 \cdot S_a \cdot e_w + 2 \cdot p \cdot (K - 0.1)} \quad \text{...mm}$$

where:

D = inside dia of spherical vessel or hemisph. head, of inside major dia for an elliptical head, or 2 * Crown radius for the ASME torisph. head.

Value of constant, K is given by:

$$K = \frac{1}{6} \cdot \left[2 + \left(\frac{D}{D_1} \right)^2 \right]$$

where: D_1 = minor dia of elliptical head, mm

For ASME torisph. head, K = 0.885

The Mathcad Function:

$$t_h(p, D, K, S_a, e_w) := \frac{p \cdot D \cdot K}{2 \cdot S_a \cdot e_w - 0.2 \cdot p} \quad \text{...mm}$$

Ex: $p := 800$ kPa $D := 3050$ mm $S_a := 129200$ kPa $e_w := 1$ $K := 0.5$

Then, we get:

$t_h(p, D, K, S_a, e_w) = 4.724$ mm .... thickness of hemisph. head of vac. jacketed tank

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6.2.5 DESIGN EQUATIONS FOR THICKNESS OF OUTER SHELL OF A DEWAR VESSEL

Outer vessel is subjected external pressure from atmosphere, and vacuum from inside. It will fail by 'elastic instability' or buckling.

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The collapsing or critical pressure for a cylinder exposed to external pressure, depends on if the cylinder is 'long' or 'short', as per the following definition:

For a **long cylinder**, following Length to Diameter ratio condition must be met:

$$L/D_o > 1.140\ (1 - \upsilon^2)^{1/4}\ (D_o/t)^{1/2}$$

where: L is the unsupported length (distance between stiffening rings for outer shell) of the cylinder.

The collapsing or critical pressure for a *long cylinder* exposed to external pressure is:

$$p_c = \frac{2 \cdot E \cdot \left(\frac{t}{D_0}\right)^3}{1 - v^2}$$

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where:

E = Young's modulus of shell material

t = shell thickness

D_0 = outside dia of shell

υ = Poisson's ratio for shell material

The collapsing or critical pressure for a short cylinder exposed to external pressure is:

$$p_c = \frac{2.42 \cdot E \cdot \left(\frac{t}{D_0}\right)^{\frac{5}{2}}}{\left(1 - v^2\right)^{\frac{3}{4}} \cdot \left[\left(\frac{L}{D_0}\right) - 0.45 \cdot \left(\frac{t}{D_0}\right)^{\frac{1}{2}}\right]}$$

where: L is the distance between stiffening rings for outer shell.

Heads for the outer vessel: These also must be designed for buckling.

Critical pressure for a hemispherical, elliptical or torispherical head (**or, for a spherical vessel**) is given by:

$$p_c = \frac{0.5 \cdot E \cdot \left(\frac{t_h}{R_0}\right)^2}{\left[3 \cdot \left(1 - v^2\right)\right]^{\frac{1}{2}}}$$

where: R_0 is the outer radius of spherical head or spherical vessel, the equivalent radius for the elliptical head, or the crown radius of the torispherical head.

The equivalent radius for elliptical heads is given by R0 = K1.D where D is the mahor dia and K1 is given in Table 6.6.

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Also, note that, as per ASME code, collapsing or critical pressure pc (in the above equations) is given by:

$$p_c = 4.p_a$$

where 4 is the factor of safety, p_a = external pressure (i.e. atm. pressure)

Prob.6.6 Write a Mathcad Functions for thickness of outer cylindrical shell of a cryogenic tank:

First, determine if it is a 'long cylinder' or 'short cylinder':

Test for a 'long' cylinder:

$$L/D_o > 1.140\ (1 - \upsilon^2)^{1/4}\ (D_o/t)^{1/2}$$

Mathcad Function to determine if it is a long or short cylinder:

$$\text{Test\_for\_long\_cyl}(L, D_0, t, \nu) := \left| \begin{array}{l} \text{LbyD0} \leftarrow \dfrac{L}{D_0} \\ \text{RHS} \leftarrow 1.14 \cdot \left(1 - \nu^2\right)^{\frac{1}{4}} \cdot \left(\dfrac{D_0}{t}\right)^{\frac{1}{2}} \\ \text{BB} \leftarrow \text{"Long cylinder!"} \quad \text{if LbyD0} \geq \text{RHS} \\ \text{CC} \leftarrow \text{"Short cylinder!"} \quad \text{if LbyD0} < \text{RHS} \\ \begin{bmatrix} \text{"L/D0"} & \text{"RHS"} & \text{"BB"} & \text{"CC"} \\ \text{LbyD0} & \text{RHS} & \text{BB} & \text{CC} \end{bmatrix} \end{array} \right.$$

Ex: $L := 2.25\ \text{m} \quad D_0 := 3.66\ \text{m} \quad t := 0.012\ \text{m} \quad \nu := 0.27$

$$\text{Test\_for\_long\_cyl}(L, D_0, t, \nu) = \begin{bmatrix} \text{"L/D0"} & \text{"RHS"} & \text{"BB"} & \text{"CC"} \\ 0.615 & 19.536 & 0 & \text{"Short cylinder!"} \end{bmatrix}$$

For Long cylinder, thickness of outer shell is given by:

Failure by Elastic instability:

p_c = collapsing pressure = 4 * p_a, p_a = atm. pressure, 4 is the factor of safety

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$$t\_outershell\_longcyl(L, D_0, p_c, E, v) := \left[\frac{p_c \cdot (1 - v^2)}{2 \cdot E}\right]^{\frac{1}{3}} \cdot D_0$$ ...for long cylinder

For short cylinder, thickness of outer shell is calculated from:

$$p_c = \frac{2.42 \cdot E \cdot \left(\frac{t}{d_0}\right)^{\frac{5}{2}}}{(1 - v^2)^{\frac{3}{4}} \cdot \left[\left(\frac{L}{D_0}\right) - 0.45 \cdot \left(\frac{t}{D_0}\right)^{\frac{1}{2}}\right]}$$

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Mathcad Function for thickness of outer shell of short cylinder:

$t := 0.001$ m .... guess value

Given

$$p_c = \frac{2.42 \cdot E \cdot \left(\frac{t}{D_0}\right)^{\frac{5}{2}}}{\left(1 - v^2\right)^{\frac{3}{4}} \cdot \left[\left(\frac{L}{D_0}\right) - 0.45 \cdot \left(\frac{t}{D_0}\right)^{\frac{1}{2}}\right]}$$

$t\_outershell\_shortcyl\left(L, D_0, p_c, E, \upsilon\right) := \text{Find}(t)$

Ex: $p_c := 405.2$ kPa $v := 0.27$ $E := 200 \cdot 10^6$ kPa $L := 2.25$ m $D_0 := 3.66$ m

Then:

$t\_outershell\_shortcyl\left(L, D_0, p_c, E, v\right) = 0.01074$ m

==========

Required head thickness for outer cylinder:

For hemispherical, elliptical or torisph. head, (or for a spherical vessel), thickness is given by:

$$t\_head\_outer\_vessel\left(p_c, E, R_0, v\right) := \left[\frac{p_c \cdot \left[3 \cdot \left(1 - v^2\right)\right]^{\frac{1}{2}}}{0.5 \cdot E} \cdot R_0^{\,2}\right]^{\frac{1}{2}}$$

where:

R_0 = outside rad. of spherical head or spherical vessel, equivalent radius of elliptical head, or crown rad. of torisph. head.

Equiv. radius (K1) for Elliptical heads:

D = Major dia, D1 = Minor dia,

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Equiv. rad. = K1. D; factor K1 is given below:

	D/D1	K1
	3.0	1.36
	2.8	1.27
	2.6	1.18
	2.4	1.08
	2.2	0.99
MM :=	2.0	0.90
	1.8	0.81
	1.6	0.73
	1.4	0.65
	1.2	0.57
	1.0	0.50

Then, K1 is determined from the following Mathcad Function:

$$K1(DbyD1) := \text{linterp}\left(\text{sort}\left(MM^{\langle 0 \rangle}\right), \text{sort}\left(MM^{\langle 1 \rangle}\right), DbyD1\right)$$

Ex:

$$K1(2.9) = 1.315$$

Ex. for head thickness calculation:

Ex: $p_c := 405.2$ kPa $\quad \nu := 0.27 \quad E := 200 \cdot 10^6$ kPa $\quad L := 2.25$ m $\quad R_0 := \frac{3.66}{2}$ m

And:

$$t\_head\_outer\_vessel\left(p_c, E, R_0, \nu\right) = 4.757 \cdot 10^{-3} \quad \text{m}$$

==========

Prob.6.7 Determine the inner vessel and outer vessel minimum thicknesses for a spherical LH2 vessel having a nominal capacity of 2000 m^3 with a 10% ullage volume. The outer shell O.D. is 1.22 m larger than the inner shell O.D. The inner shell is constructed of 304 SS with spot examined butt joints with complete penetration welds. The outer shell is constructed of carbon steel - SA 285, grade C. The design pressure is 700 kPa absolute. [1]

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Solution:

Inner vessel:

$V_{inner} := 2000 + 0.1 \cdot 2000$ m^3

i.e. $V_{inner} = 2.2 \cdot 10^3$ m^3

$$R_{inner} := \left(V_{inner} \cdot \frac{3}{4} \cdot \frac{1}{\pi}\right)^{\frac{1}{3}}$$

i.e. $R_{inner} = 8.068$ m

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And, $D_{inner} := 2 \cdot R_{inner}$

i.e. $D_{inner} = 16.136$ m.... inner shell dia

Therefore: $D_{outer} := D_{inner} + 1.22$

i.e. $D_{outer} = 17.356$ m .... outer shell dia

Thickness of inner spherical shell:

Allowable stress for SS 304: See Table 6.1

Or, using the Mathcad Function already written:

MaterialNumber := 8

$$Allowable\_stress(MaterialNumber) = \begin{bmatrix} \text{"MaterialNumber"} & \text{"Material"} & \text{"Allowable stress, S\_a (MPa)"} \\ 8 & \text{"Stainless steel SA-240 (304)"} & 129.2 \end{bmatrix}$$

i.e. $S_a := 129.2 \cdot 10^3$ kPa

We have, for sph. shell:

$$t_h(p, D, K, S_a, e_w) := \frac{p \cdot D \cdot K}{2 \cdot S_a \cdot e_w - 0.2 \cdot p} \quad \text{...mm}$$

We have:

$p := 700$ kPa $D := D_{inner}$ m $S_a := 129200$ kPa

$e_w := 0.85$ ..for spot examined butt joints $K := 0.5$ ...for sph. head

Then: $t_{innershell} := t_h(p, D, K, S_a, e_w)$

i.e. $t_{innershell} = 0.02573$ **m ... Ans.**

Now, tolerance for metric plates is 0.3 mm under the standard thickness.

So, the nominal thickness would be (26 mm + 0.3 mm) = 26.3 mm.

Therefore, select 28 mm plate; see Table 6.4

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i.e. choose 28 mm (1 1/8 in.) plate for inner spherical shell ..... Ans.

Therefore, OD of innershell: $OD_{inner} := D_{inner} + 2 \cdot 0.028$ m

i.e. $OD_{inner} = 16.192$ m

Then, OD of outer sph. shell: $OD_{outershell} := OD_{inner} + 1.22$

i.e. $OD_{outershell} = 17.412$ m .... outer shell dia

Thickness of outer spherical shell:

$p_c := 4 \cdot 101.3$ kPa ... critical (or collapsing pressure), 4 is factor of safety, 101.3 is atm. preassure in kPa

For carbon steel SA 285-grade C, Allowable stress is given by the Mathcad function written earlier (or, from Table 6.1):

Using the Mathcad function:

MaterialNumber := 0

Allowable_stress(MaterialNumber) =

"MaterialNumber"	"Material"	"Allowable stress, S_a (MPa)"
0	"Carbon steel SA-285 Grade C"	94.8

i.e. allowable stress = 94.8 MPa

Young's Modulus for the material; using the Mathcad function already written:

MaterialNumber := 0

Youngs_Modulus_etc(MaterialNumber) =

"MaterialNumber"	"Material"	"Density, rho (kg/m^3)"	"Young's Mod., E_y (GPa)"	"Poisson's ratio, nu"
0	"Carbon steel"	$7.72 \cdot 10^3$	200	0.27

Therefore: $E := 200 \cdot 10^6$ kPa .... Young's Mod.

$\nu := 0.27$ ...Poisson's ratio

$R_0 := \frac{OD_{outershell}}{2}$ ...radius of outer shell

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Then, we have:

$$t\_head\_outer\_vessel(p_c, E, R_0, \nu) := \left[\frac{p_c \cdot \left[3 \cdot \left(1 - \nu^2 \right) \right]^{\frac{1}{2}}}{0.5 \cdot E} \cdot R_0^{\,2} \right]^{\frac{1}{2}}$$

where:

R_0 = outside rad. of spherical head or spherical vessel, equivalent radius of elliptical head, or crown rad. of torisph. head.

$$t_{h\_outer} := t\_head\_outer\_vessel(p_c, E, R_0, \nu)$$

i.e. $t_{h\_outer} = 0.023$ m

i.e. choose 25 mm (1 in.) plate for outer spherical shell ..... Ans.

==========

Prob.6.8. The inner shell of a horizontal cylindrical liquid-nitrogen dewar has a cylindrical length of 24 m (78.74 ft) and an inside diameter of 1.829 m (72 in.). The inner vessel is constructed of 304 stainless steel with 100 percent radio-graphed welds. The working pressure for the inner vessel is 620 kPa (absolute) or 89.9 psia. Elliptical heads with D/ D1 = 2 are used as the end closures for the vessel. Determine: the minimum thickness of the inner shell and heads. The ullage volume is 10 percent. [1]

Solution:

Inner vessel design:

Material: SS 304, 100% radiographed welds.

$L := 24$ m ... length of cyl. section

$D_{inner} := 1.829$ m ... inside dia of cyl. shell

$p := 620$ kPa ... inside pressure

Two elliptical headfs with D/D1 = 2

Therefore, vol. of cyl. section:

$$V_{cyl} := \frac{\pi \cdot D_{inner}^{\,2}}{4} \cdot L \qquad \text{i.e.} \quad V_{cyl} = 63.056 \quad \text{m\^3}$$

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Vol. of two elliptical heads:

$$V_{heads} := 2 \cdot \frac{\pi \cdot D_{inner}^{3}}{24} \qquad \text{i.e.} \quad V_{heads} = 1.602 \quad m\text{^}3$$

Therefore, total inner cyl. volume:

$$V_{inner} := V_{cyl} + V_{heads} \quad m\text{^}3$$

i.e. $V_{inner} = 64.658$ m^3

Including 10% ullage volume, we get:

$$V_{inner} := V_{inner} + 0.1 \cdot V_{inner}$$

i.e. $V_{inner} = 71.124$ **m^3 .... total inner cyl vol.**

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Thickness of inner cyl. shell:

Allowable stress for SS 304:

MaterialNumber := 8

$$\text{Allowable\_stress}(\text{MaterialNumber}) = \begin{bmatrix} \text{"MaterialNumber"} & \text{"Material"} & \text{"Allowable stress, S\_a (MPa)"} \\ 8 & \text{"Stainless steel SA-240 (304)"} & 129.2 \end{bmatrix}$$

i.e. $S_a := 129.2 \cdot 10^3$ kPa

Now, we have:

$$t_{inner\_shell}(p, D, S_a, e_w) := \frac{p \cdot D}{2 \cdot S_a \cdot e_w - 1.2 \cdot p} \quad \text{... mm}$$

And: $p := 620$ kPa $D := 1829$ mm $S_a := 129200$ kPa

$e_w := 1$ ...for 100% radiographed welds

Then, we get:

$t_{inner\_shell}(p, D, S_a, e_w) = 4.401$ mm .... thickness of cyl. inner shell of vac. jacketed tank

i.e. choose 5 mm (3/16 in.) plate for inner cylindrical shell ..... Ans.

Thickness of elliptical heads of inner vessel:

Min. thickness for spherical shells, hemisph. heads, Elliptical heads, or ASME torispherical heads is given by:

$$t_h = \frac{p \cdot D \cdot K}{2 \cdot S_a \cdot e_w - 0.2 \cdot p} = \frac{p \cdot D_0 \cdot K}{2 \cdot S_a \cdot e_w + 2 \cdot p \cdot (K - 0.1)} \quad \text{...mm}$$

where:

D = inside dia of spherical vessel or hemisph. head, of inside major dia for an elliptical head, or 2 * Crown radius for the ASME torisph. head.

Value of constant, K is given by:

$$K = \frac{1}{6} \cdot \left[2 + \left(\frac{D}{D_1} \right)^2 \right]$$

where: D1 = minor dia of elliptical head, mm

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For ASME torisph. head, K = 0.885

For D/D1 = 2, we get: $K := 1$

Therefore, we get:

$$t_h := \frac{p \cdot D \cdot K}{2 \cdot S_a \cdot e_w - 0.2 \cdot p}$$

i.e. $t_h = 4.391$ mm ... thickness of elliptical head

i.e. Again, choose 5 mm (3/16 in.) plate for inner shell elliptical heads ..... Ans.

Prob. 6.9 The outer shell of a liquid-nitrogen storage vessel has a cylindrical length of 24.8 m (81.36 ft) and an outside diameter of 2.438 m (96 in.). The outer vessel is constructed of carbon steel, SA-285, grade C. Elliptical heads with D/ D1 = 2 are used as the end closures for the vessel. Eight stiffening rings (two main support rings and six intermediate rings) spaced 2.75 m (108.3 in.) apart are used on the outer shell. Determine: the minimum thickness of the outer shell and heads. [1]

Solution:

This is part of a problem from Ref. [1].

For Carbon steel, from Table 6.5, we have:

$E := 200 \cdot 10^6$ kPa ..... Youngs modulus

$\nu := 0.27$ .....Poissons Ratio

$L := 2.75$ m ... distance between stiffening rings

$D_o := 2.438$ m

$p_c := 4 \cdot 101.3$ kPa

For Elliptical heads with D/D1 = 2, we have from Table 6.6:

$K1 := 0.9$

And: $R_o = K1 \cdot D$ where Ro is the equiv. radius for elliptical heads under external pressure

D1 = minor dia, and D = major dia of the elliptical head

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First, assume that it is a short cylinder and find out the shell thickness. Then, verify, applying the test for a long cylinder, that it is really a short cylinder:

Using the Mathcad Function already written, for a short cylinder:

$$t\_outershell\_shortcyl(L, D_o, p_c, E, v) = 9.182 \cdot 10^{-3} \quad m$$

i.e. shell thickness is : 9.182 mm.... choose10 mm nominal plate, from Table 6.4

Now, apply the test to see if it is really a short cyl or not:

$$t := 9.182 \cdot 10^{-3} \quad m$$

$$Test\_for\_long\_cyl(L, D_o, t, v) = \begin{bmatrix} \text{"L/D0"} & \text{"RHS"} & \text{"BB"} & \text{"CC"} \\ 1.128 & 18.228 & 0 & \text{"Short cylinder!"} \end{bmatrix}$$

Therefore, confirmed that it is a short cylinder.

Thickness of elliptical heads:

We have for elliptical head thickness:

$$t\_head\_outer\_vessel(p_c, E, R_0, v) := \left[\frac{p_c \cdot \left[3 \cdot \left(1 - v^2\right)\right]^{\frac{1}{2}}}{0.5 \cdot E} \cdot R_0^{\,2} \right]^{\frac{1}{2}}$$

where:

R_0 = outside rad. of spherical head or spherical vessel, equivalent radius of elliptical head, or crown rad. of torisph. head.

$$R_0 := K1 \cdot D_o \qquad R_0 = 2.194 \quad m$$

Then, elliptical head thickness:

$$t\_head\_outer\_vessel(p_c, E, R_0, v) = 5.704 \cdot 10^{-3} \quad m$$

i.e. head thickness = 5.704 mm ...i.e. use 6 mm thick head .... Ans.

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6.2.6 DESIGN EQUATIONS FOR PIPING

Piping is required to remove liquid from the container, to vent vapor etc. It also introduces a heat leak into the product container, and should be carefully designed. Bellows are used to take care of thermal contraction. Also, a vertical rise is used in the pipe in the vacuum space to prevent convection problems. Four piping arrangements are shown below [1]:

Arrangement (4) is the best arrangement, since the bellows is located in the stand-off and is easy to repair, if the bellows develops a leak by fatigue crack.

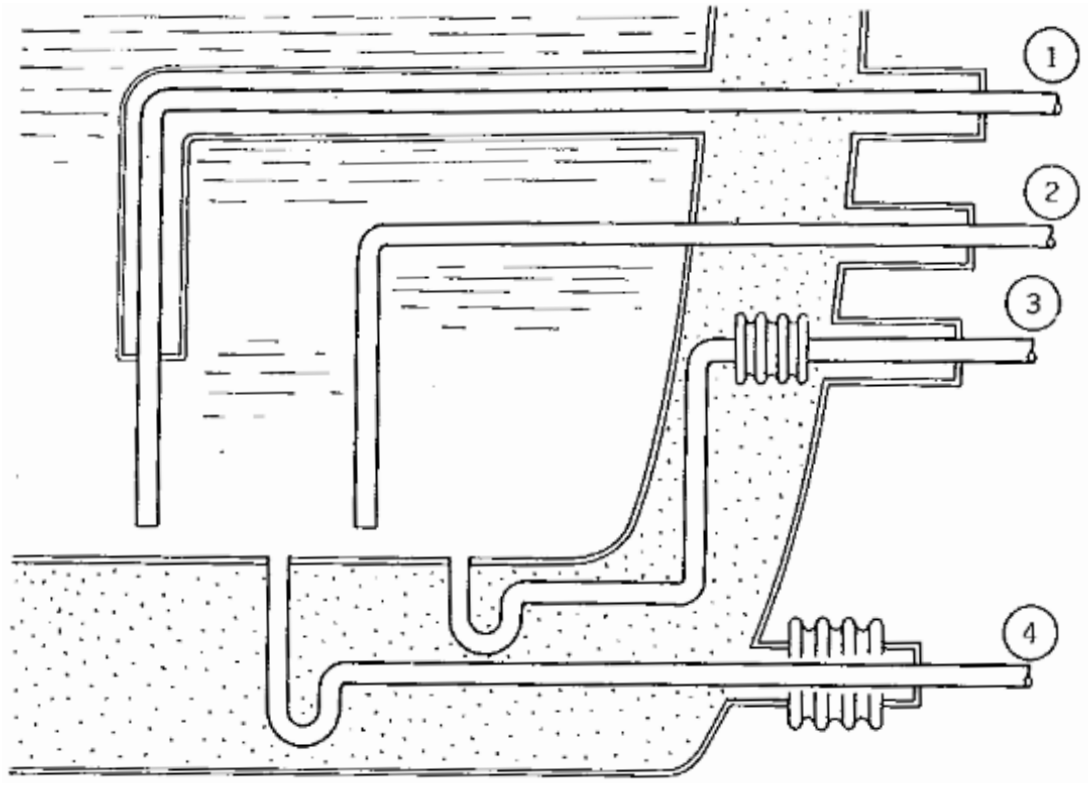

Fig 6.2 Cryogenic fluid storage vessel – piping arrangements.

The minimum wall thickness for piping subjected to internal pressure is determined from:

$$t = p \cdot \frac{D_o}{2 \cdot s_a + 0.8 \cdot p}$$

where: p = design pressure, Do = outside dia of pipe, sa = allowable stress of pipe material.

For piping subjected to external pressure, thickness is calculated from the formula used for long cylinder subjected to external pressure, viz.

Standard dimensions for pipe sizes are given in Tables 6.8 and 6.9.

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6.2.7 HEAT TRANSFER FROM PIPING OR SUPPORT RODS

Considering variable thermal conductivity of material with temperature, following formula gives the heat transfer rate through the piping (or support rod), as it passes from the high temp region to low temp region:

$$Q=\left(K_h - K_c\right)\cdot\left(\frac{A}{L}\right)$$ where:

Th = temp at warm end

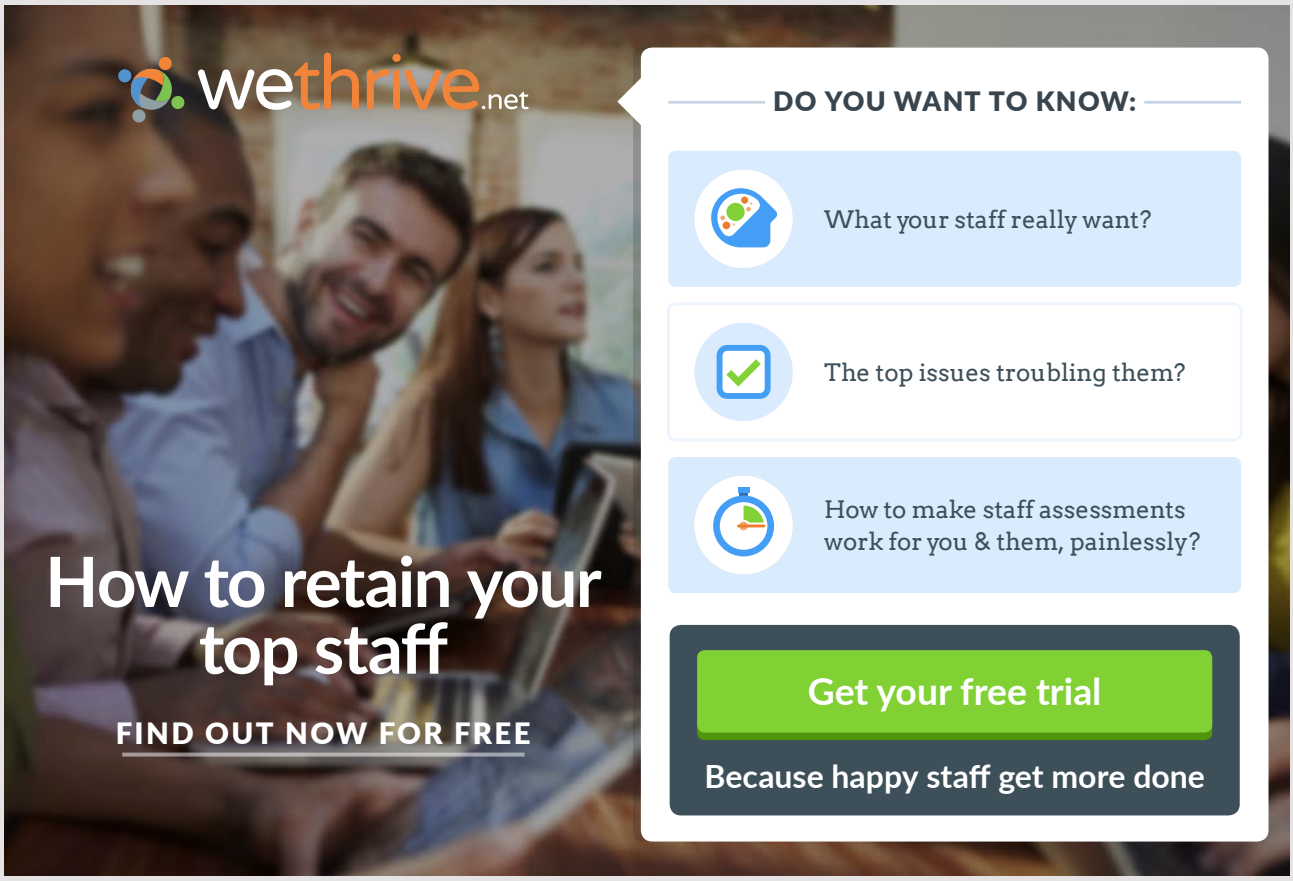
Tc = temp at cold end

A = cross-sectional area of rod

L = length of rod, and

$$K=\int_{4\,K}^{T} k_t\,dT$$ = thermal conductivity integral.... from 4K to T

Values of thermal conductivity and thermal cond. integrals for several materials are given in Table 6.7 [2]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6.2.8 MATHCAD FUNCTIONS FOR THERMAL CONDUCTIVITY INTEGRALS

It will be quite useful to write Mathcad Function to find Thermal conductivity integral for a given material when it is cooled from a warm temperature TH to a cold temperature TL.

To do this, let us first write the data from Table 6.7 for some materials in a Matrix, MM as shown below.

In this Matrix, first column gives Temperatures in Kelvin and other columns give Thermal conductivity integrals for different materials between a given high temperature, T and low temperature of 4 K.

$MM :=$

T (K)	BeCu	AL 6063	Steel 1020	SS 304	Monel	Teflon
4	0	0	0	0	0	0
10	19	360	43	2.9	6.3	0.44
20	95	1650	222	16.3	36.4	1.64
30	229	3650	502	42.4	92.9	3.23
40	415	6200	867	82.4	173	5.08
50	650	8950	1310	135	273	7.16
60	930	11700	1810	198	368	9.36
70	1250	14300	2360	270	513	11.6
80	1600	16700	2950	349	647	13.9
90	1990	19000	3550	436	791	16.3
100	2400	21100	4170	528	940	18.7
120	3300	25300	5450	726	1260	23.7
140	4320	29300	6750	939	1590	28.7
160	5440	33300	8050	1170	1950	33.8
180	6640	37300	9350	1410	2320	39
200	7910	41300	10700	1660	2710	44.2
250	11300	51300	13900	2340	3730	57.2
300	15000	61300	17200	3060	4800	70.2

$Temp := MM^{\langle 0 \rangle}$

$Temp =$

	0
0	4
1	10
2	20
3	30
4	40
5	50
6	60
7	70
8	80
9	90
10	100
11	120
12	140
13	160
14	180
15	200
16	250
17	300

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One example of Thermal conductivity integral for BeCu from T to 4K is shown below:

Th. cond. Integral for BeCU:

$$\text{ThCond\_Integral\_Becu\_T\_to\_4K}(T) := \begin{cases} \text{return "T must be between 4 K and 300 K!"} & \text{if } T < 4 \\ \text{return "T must be between 4 K and 300 K!"} & \text{if } T > 300 \\ \text{linterp}\left[\text{Temp}, (MM)^{\langle 1 \rangle}, T\right] \end{cases}$$

Ex: $\text{ThCond\_Integral\_Becu\_T\_to\_4K}(190) = 7.275 \cdot 10^{3}$ W/m

---

Generally, however, we are interested in Thermal conductivity integrals between the warm end at a temp. of TH and the cold end at a temp of TL, so that we can calculate the heat transferred by conduction from warm end to cold end. So, we write useful functions for a few important materials below:

$$\text{ThCond\_Integral\_BeCu\_TH\_to\_TL}(TH, TL) := \begin{cases} \text{return "TL, TH must be between 4 K and 300 K!"} & \text{if } TL < 4 \\ \text{return "TL, TH must be between 4 K and 300 K!"} & \text{if } TH > 300 \\ \text{linterp}\left[\text{Temp}, (MM)^{\langle 1 \rangle}, TH\right] - \text{linterp}\left[\text{Temp}, (MM)^{\langle 1 \rangle}, TL\right] \end{cases}$$

Ex: $\text{ThCond\_Integral\_BeCu\_TH\_to\_TL}(300, 80) = 1.34 \cdot 10^{4}$ W/m

---

$$\text{ThCond\_Integral\_AL6063\_TH\_to\_TL}(TH, TL) := \begin{cases} \text{return "TL, TH must be between 4 K and 300 K!"} & \text{if } TL < 4 \\ \text{return "TL, TH must be between 4 K and 300 K!"} & \text{if } TH > 300 \\ \text{linterp}\left[\text{Temp}, (MM)^{\langle 2 \rangle}, TH\right] - \text{linterp}\left[\text{Temp}, (MM)^{\langle 2 \rangle}, TL\right] \end{cases}$$

Ex: $\text{ThCond\_Integral\_AL6063\_TH\_to\_TL}(300, 80) = 4.46 \cdot 10^{4}$ W/m

---

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$$\text{ThCond\_Integral\_Steel1020\_TH\_to\_TL}(TH, TL) := \begin{vmatrix} \text{return "TL, TH must be between 4 K and 300 K!"} & \text{if } TL < 4 \\ \text{return "TL, TH must be between 4 K and 300 K!"} & \text{if } TH > 300 \\ \text{linterp}\left[\text{Temp}, (MM)^{\langle 3 \rangle}, TH\right] - \text{linterp}\left[\text{Temp}, (MM)^{\langle 3 \rangle}, TL\right] \end{vmatrix}$$

Ex: $\text{ThCond\_Integral\_Steel1020\_TH\_to\_TL}(300, 80) = 1.425 \cdot 10^4$ W/m

---

$$\text{ThCond\_Integral\_SS304\_TH\_to\_TL}(TH, TL) := \begin{vmatrix} \text{return "TL, TH must be between 4 K and 300 K!"} & \text{if } TL < 4 \\ \text{return "TL, TH must be between 4 K and 300 K!"} & \text{if } TH > 300 \\ \text{linterp}\left[\text{Temp}, (MM)^{\langle 4 \rangle}, TH\right] - \text{linterp}\left[\text{Temp}, (MM)^{\langle 4 \rangle}, TL\right] \end{vmatrix}$$

Ex: $\text{ThCond\_Integral\_SS304\_TH\_to\_TL}(77, 4) = 325.3$ W/m

---

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$$\text{ThCond\_Integral\_Monel\_TH\_to\_TL}(TH, TL) := \begin{cases} \text{return "TL, TH must be between 4 K and 300 K!" if } TL < 4 \\ \text{return "TL, TH must be between 4 K and 300 K!" if } TH > 300 \\ \text{linterp}\left[\text{Temp}, (MM)^{\langle 5 \rangle}, TH\right] - \text{linterp}\left[\text{Temp}, (MM)^{\langle 5 \rangle}, TL\right] \end{cases}$$

Ex: $\text{ThCond\_Integral\_Monel\_TH\_to\_TL}(300, 80) = 4.153 \cdot 10^3$ W/m

---

$$\text{ThCond\_Integral\_Teflon\_TH\_to\_TL}(TH, TL) := \begin{cases} \text{return "TL, TH must be between 4 K and 300 K!" if } TL < 4 \\ \text{return "TL, TH must be between 4 K and 300 K!" if } TH > 300 \\ \text{linterp}\left[\text{Temp}, (MM)^{\langle 6 \rangle}, TH\right] - \text{linterp}\left[\text{Temp}, (MM)^{\langle 6 \rangle}, TL\right] \end{cases}$$

Ex: $\text{ThCond\_Integral\_Teflon\_TH\_to\_TL}(300, 80) = 56.3$ W/m

---

Note: Many useful Tables/graphs for sp. heats, thermal conductivities, and thermal conductivity integrals for materials used in cryogenic engineering are given in Ref.[8].

6.2.9 DRAINING THE VESSEL

There are 3 methods: (a) Self – pressurization of the inner vessel – here, a part of the liquid from the inner vessel is removed and boiled outside the vessel in an evaporator and resulting high volume vapor is passed back to the ullage space, causing a pressure rise. (b) External gas pressurization ---- here, a high pressure gas from an external source is passed in to the container; part of the liquid may be pumped to a high pressure and then vaporized electrically and stored in gas cylinders for use when needed. (c) Use of a cryogenic pump in the drain line to remove the liquid --- this is useful when large flow rates are desired.

To determine the gas mass required in pressurizing the storage vessel:

For transfer of liquid Oxygen, liquid Argon and liquid Nitrogen with a condensable pressurizing gas, the gas requirement is determined from the saturation rule, as follows:

$$\Delta m_g = V2 \cdot \rho_{g2} - V1 \cdot \rho_{g1}$$

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where:

V1 = initial ullage volume

V2 = final ullage volume

ρ_{g1} = sat. vapor density at the initial pressure

ρ_{g2} = sat. vapor density at the final pressure

Note: For liquid hydrogen pressurization, more complicated relations are available. Reader may refer to Ref. [1]

6.2.10 SAFETY DEVICES [1]

Basic minimum safety devices used in large cryogenic storage vessels are: (a) inner vessel pressure relief valve, (b) inner shell burst-disc assembly, and (c) annular space burst-disc assembly.

Location of safety devices in a dewar is shown in the following fig.:

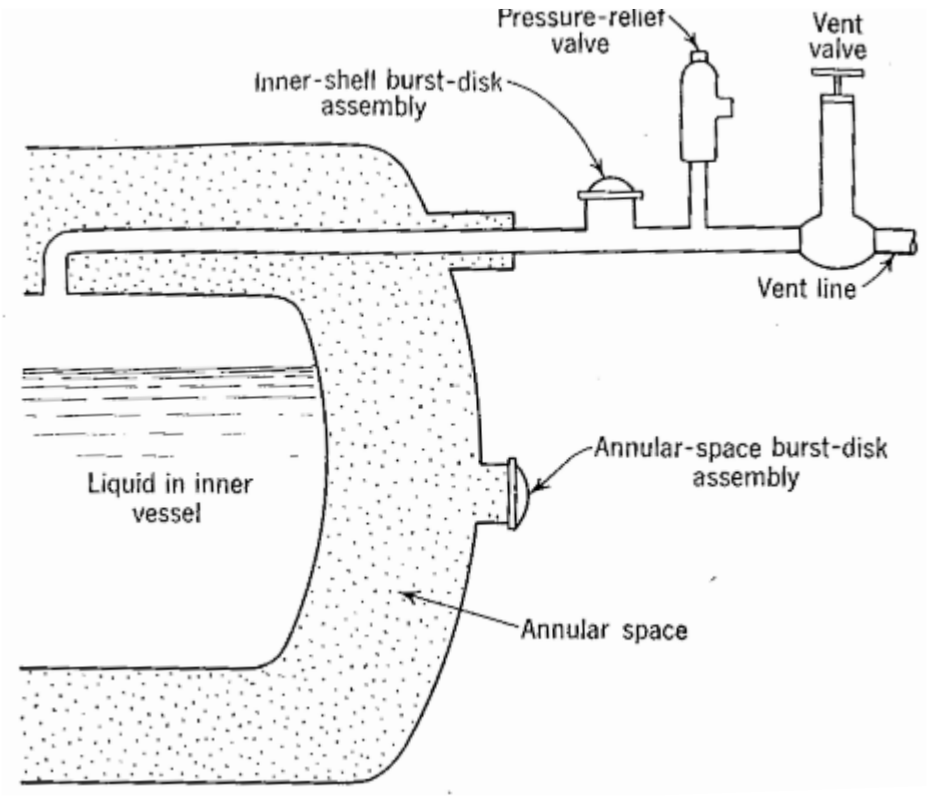

Fig 6.3 Location of basic safety devices in a dewar.

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Inner vessel pressure relief valve is generally spring loaded type. Used to prevent excessive pressure build up in inner vessel, if the annular space vacuum is lost.

The required size of safety valve is determined by ASME code as:

$$A_v = \frac{m_g \cdot \left(\frac{R_u \cdot T}{M}\right)^{\frac{1}{2}}}{C \cdot K_D \cdot P_{max}}$$

where

A_v = discharge area of valve

m_g = max. mass flow rate through the valve

R_u = universal gas constant = 8.314 kg/kmol.K

T = abs. temp of gas at the inlet to the valve

M = Mol. wt. of gas flowing

K_D = discharge coeff., determined by test

p_{max} = (set gauge pressure * 1.1) + atm. pressure

$$C = \left[\gamma \cdot \left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma+1}{\gamma-1}}\right]^{\frac{1}{2}}$$

$\gamma = \frac{c_p}{c_v}$ = ratio of sp. heats

Prob. 6.10 The fill-and drain line in a liquid oxygen vessel is a 100-mm nominal (4 in. nominal) stainless-steel pipe. The design pressure of the vessel is 860 kPa (absolute) or 124.7 psia, and the line passes through the vacuum space for a distance of 7.50 m (24.6 ft). The warm end of the line is at 300 K (540°R) and the cold end is at 90 K (162°R), and no boiling-condensation takes place within the line. Determine: (a) the minimum, thickness and schedule number required; and (b) the heat conducted down the line to the inner vessel [1].

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Solution:

For thickness of the pipe subjected to internal pressure, we have:

$$t = p \cdot \frac{D_o}{2 \cdot s_a + 0.8 \cdot p}$$ ....thickness for piping

where: p = design pressure, Do = outside dia of pipe, sa = allowable stress of pipe material.

Now: $p := 860$ kPa

$D_o := 114.3$ mm .... from Table 6.9

$s_a := 129200$ kPa ...for SS 304 ... from Table 6.1

Therefore:

$$t := p \cdot \frac{D_o}{2 \cdot s_a + 0.8 \cdot p}$$

i.e. $t = 0.3794$ mm... thickness of pipe

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But, 100 mm nominal pipe, sch. 5 has a thickness of 2.1 mm. So, adopt the same.

Heat conducted down the line length of 7.5 m, from 300 K to 90 K:

$L := 7.5$ m

$D_i := D_o - 2.1 \cdot 2$ i.e. $D_i = 110.1$ mm

$A := \pi \cdot \frac{\left(D_o^2 - D_i^2\right)}{4} \cdot 10^{-6}$ m^2..... area of cross-section of pipe

i.e. $A = 7.402 \cdot 10^{-4}$ m^2

Then, considering the variable thermal conductivity of the material with temp:

$$Q = \left(K_h - K_c\right) \cdot \left(\frac{A}{L}\right)$$ where:

Th = temp at warm end

Tc = temp at cold end

A = cross-sectional area of rod

L = length of rod, and

$K = \int_{4\,K}^{T} k_t dT$ = thermal conductivity integral.... from 4K to T

Values of thermal conductivity and thermal cond. integrals for several materials are given in Table 6.7

We get, from Table 6.7::

$K_h := 3060$ W/m .... at 300 K

$K_c := 436$ W/m .. at 90 K

And, $K_h - K_c = 2624$ W/m

And:

$$Q := \left(K_h - K_c\right) \cdot \left(\frac{A}{L}\right)$$

i.e. $Q = 0.25898$ **W .....Ans.**

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OR:

Using the Mathcad Function written earlier, we can get the Th. cond. integral for SS304 between 300K and 90K:

$$\text{ThCond\_Integral\_SS304\_TH\_to\_TL}(TH, TL) := \begin{cases} \text{return "TL, TH must be between 4 K and 300 K!"} & \text{if } TL < 4 \\ \text{return "TL, TH must be between 4 K and 300 K!"} & \text{if } TH > 300 \\ \text{linterp}\left[\text{Temp}, (MM)^{\langle 4 \rangle}, TH\right] - \text{linterp}\left[\text{Temp}, (MM)^{\langle 4 \rangle}, TL\right] \end{cases}$$

i.e. $\text{ThCond\_Integral\_SS304\_TH\_to\_TL}(300, 90) = 2.624 \cdot 10^{3}$ W/m

---

Alternatively:

Let us get the equation for thermal conductivity of SS304 from NIST, See Ref [14].

https://trc.nist.gov/cryogenics/materials/304Stainless/304Stainless_rev.htm

	Thermal Conductivity	Specific Heat
UNITS	W/(m-K)	J/(kg-K)
a	-1.4087	22.0061
b	1.3982	-127.5528
c	0.2543	303.647
d	-0.6260	-381.0098
e	0.2334	274.0328
f	0.4256	-112.9212
g	-0.4658	24.7593
h	0.1650	-2.239153
i	-0.0199	0
data range	4-300	4-300
equation range	1-300	4-300
curve fit % error relative to data	2	5

And, the equation given is:

Curve fit equation of the form:

$$\log_{10} y = a + b(\log_{10} T) + c(\log_{10} T)^2 + d(\log_{10} T)^3 + e(\log_{10} T)^4 + f(\log_{10} T)^5 + g(\log_{10} T)^6 + h(\log_{10} T)^7 + i(\log_{10} T)^8$$

Solves as:

$$y = 10^{a + b(\log_{10} T) + c(\log_{10} T)^2 + d(\log_{10} T)^3 + e(\log_{10} T)^4 + f(\log_{10} T)^5 + g(\log_{10} T)^6 + h(\log_{10} T)^7 + i(\log_{10} T)^8}$$

Where: Coefficients *a* - *i* are summarized in the appropriate table and T is the temperature in K (x-axis), and y is the property to solve for.

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Let us write a Mathcad Function for k_t (W/m.K) as a function of T, using the above coefficients:

```
Th_cond_SS304(T) := | return "T should be between 4 K and 300 K!!"  if T < 4
                    | return "T should be between 4 K and 300 K!!"  if T > 300
                    | AA ← -1.4087 + 1.3982·log(T) + 0.2543·log(T)^2 - 0.626·log(T)^3 + 0.2334·log(T)^4 + 0.4256·log(T)^5 - 0.4658·log(T)^6 + 0.1650·log(T)^7 - 0.0199·log(T)^8
                    | 10^AA
```

As an example, for 150 K, k_t is:

Ex: T := 160 K

Th_cond_SS304(T) = 11.4791 W/m.K

Now, Integrate directly in Mathcad, using the equation for thermal conductivity k_t(T), from90 K to 300 K:

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Then

$$\int_{90}^{300} \text{Th\_cond\_SS304}(T)\,dT = 2.5965 \cdot 10^{3} \quad \text{W/m}$$

And:

$L := 7.5$ m $\quad A := 7.402 \cdot 10^{-4}$ m^2

Therefore: $\frac{A}{L} \cdot \int_{90}^{300} \text{Th\_cond\_SS304}(T)\,dT = 0.2563$ **W.....Ans.**

Note: this result matches well with the earlier value of Q obtained.

Prob. 6.11. Using the saturation rule, determine the pressurization mass requirement for a 135 m3 (35,660 gal U.S.) liquid-nitrogen storage vessel with 10 percent ullage volume. The transfer rate is 45 dm3/s (713 gal/min). The vessel must be maintained at 450 kPa (absolute) or 65.3 psia during a 20-minute transfer period. Assume that the vessel contains 135 m^3 of liquid nitrogen at 101.3 kPa (1 atm) initially, is suddenly pressurized, and transfer is carried out. [1]

Solution:

For transfer of liquid Oxygen, liquid Argon and liquid Nitrogen with a condensable pressurizing gas, the gas requirement is determined from the saturation rule, as follows:

$\Delta m_g = V2 \cdot \rho_{g2} - V1 \cdot \rho_{g1}$ where:

V1 = initial ullage volume

V2 = final ullage volume

ρ_{g1} = sat. vapor density at the initial pressure

ρ_{g2} = sat. vapor density at the final pressure

Now: $V1 := 135 \cdot 0.1$ m^3 ....ullage volume

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Volume transferred from the tank in 20 minutes:

vol_trans := $45 \cdot 60 \cdot 20 \cdot 10^{-3}$ m^3 ... since volume is transferred at a rate of 45 lit/s

i.e. vol_trans = 54 m^3

Therefore, final ullage volume, V2:

V2 := V1 + vol_trans i.e. V2 = 67.5 m^3

P1 := 101.3 kPa.....initial pressure

P2 := 450 kPa...final pressure

Then, find out the sat. vapor pressures ρg1 and ρg2 at the initial and final pressures.

We can get ρg1 and ρg2 from NIST – chemistry site (or, from the built-in functions of EES):

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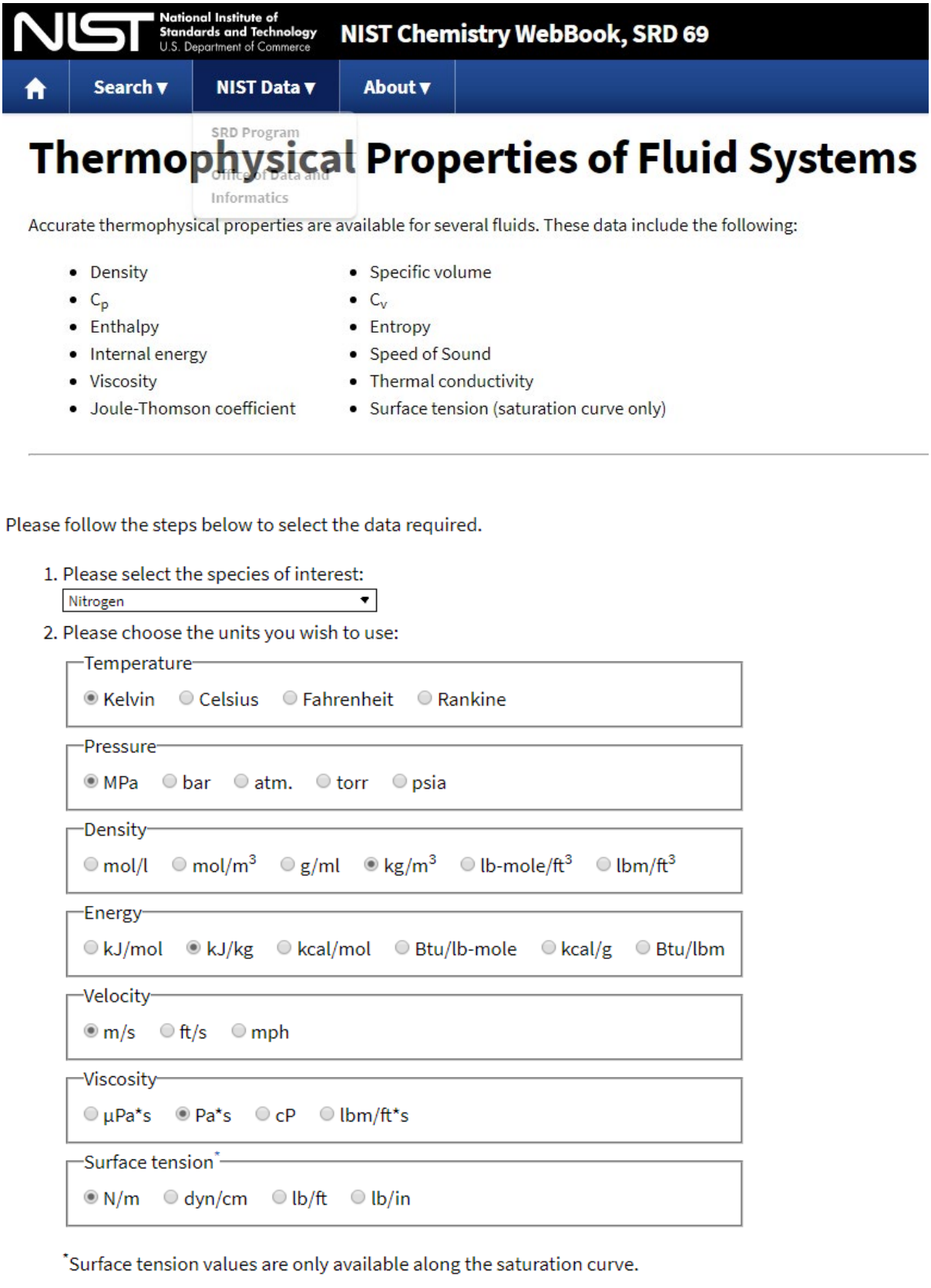
Going to the NIST site (Ref [4], i.e.: https://webbook.nist.gov/chemistry/fluid/), we get the opening screen, wherein we select the fluid (i.e. nitrogen) and the units for different properties, and the type of data desired (i.e. we require sat. properties) as shown below:



NIST National Institute of Standards and Technology U.S. Department of Commerce — NIST Chemistry WebBook, SRD 69

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Thermophysical Properties of Fluid Systems

Accurate thermophysical properties are available for several fluids. These data include the following:

- Density
- C_p
- Enthalpy
- Internal energy
- Viscosity
- Joule-Thomson coefficient
- Specific volume
- C_v
- Entropy
- Speed of Sound
- Thermal conductivity
- Surface tension (saturation curve only)

Please follow the steps below to select the data required.

1. Please select the species of interest:
   Nitrogen
2. Please choose the units you wish to use:

Temperature
◉ Kelvin ○ Celsius ○ Fahrenheit ○ Rankine

Pressure
◉ MPa ○ bar ○ atm. ○ torr ○ psia

Density
○ mol/l ○ mol/m^3 ○ g/ml ◉ kg/m^3 ○ lb-mole/ft^3 ○ lbm/ft^3

Energy
○ kJ/mol ◉ kJ/kg ○ kcal/mol ○ Btu/lb-mole ○ kcal/g ○ Btu/lbm

Velocity
◉ m/s ○ ft/s ○ mph

Viscosity
○ µPa*s ◉ Pa*s ○ cP ○ lbm/ft*s

Surface tension*
◉ N/m ○ dyn/cm ○ lb/ft ○ lb/in

*Surface tension values are only available along the saturation curve.

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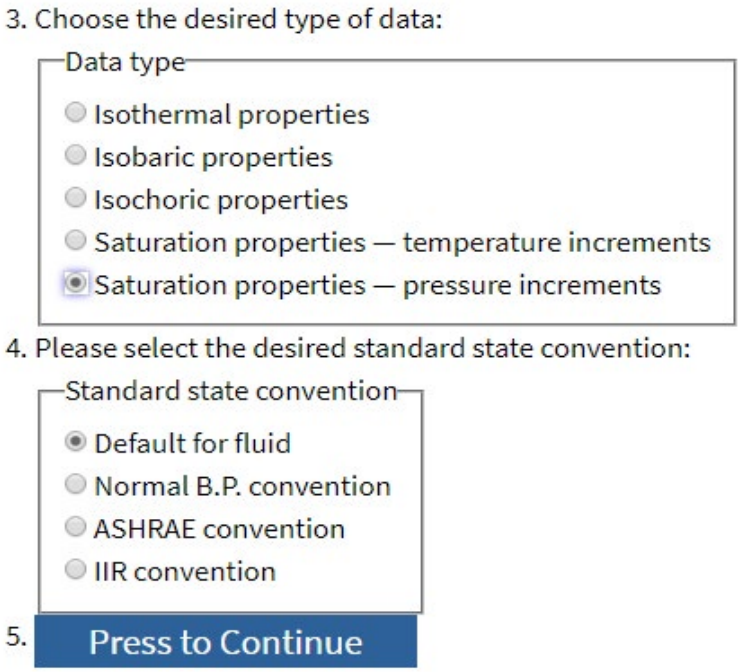
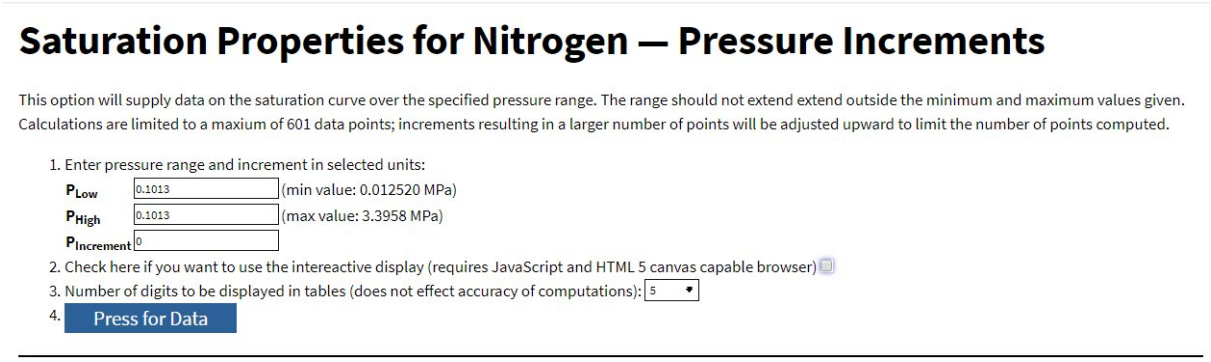


Now, press to continue, and we get:


Saturation Properties for Nitrogen — Pressure Increments
This option will supply data on the saturation curve over the specified pressure range. The range should not extend extend outside the minimum and maximum values given.
Calculations are limited to a maxium of 601 data points; increments resulting in a larger number of points will be adjusted upward to limit the number of points computed.
1. Enter pressure range and increment in selected units:
PLow 0.1013 (min value: 0.012520 MPa)
PHigh 0.1013 (max value: 3.3958 MPa)
PIncrement 0
2. Check here if you want to use the intereactive display (requires JavaScript and HTML 5 canvas capable browser)
3. Number of digits to be displayed in tables (does not effect accuracy of computations): 5
4. Press for Data

For both low and high Pressure, we have entered 0.1013 MPa, and the pressure increment as zero, since we need the sat. vap. density at 101.3 kPa only. Click on Press for Data, and we get:

Liquid Phase Data

Data on Saturation Curve

Temperature (K)	Pressure (MPa)	Density (kg/m3)	Volume (m3/kg)	Internal Energy (kJ/kg)	Enthalpy (kJ/kg)	Entropy (J/g*K)	Cv (J/g*K)	Cp (J/g*K)	Sound Spd. (m/s)	Joule-Thomson (K/MPa)	Viscosity (Pa*s)	Therm. Cond. (W/m*K)	Surf. Tension (N/m)	Phase
77.353	0.10130	806.09	0.0012406	-122.15	-122.02	2.8341	1.0841	2.0415	851.41	-0.34113	0.00016067	0.14581	0.0088753	liquid

Vapor Phase Data

Data on Saturation Curve

Temperature (K)	Pressure (MPa)	Density (kg/m3)	Volume (m3/kg)	Internal Energy (kJ/kg)	Enthalpy (kJ/kg)	Entropy (J/g*K)	Cv (J/g*K)	Cp (J/g*K)	Sound Spd. (m/s)	Joule-Thomson (K/MPa)	Viscosity (Pa*s)	Therm. Cond. (W/m*K)	Phase
77.353	0.10130	4.6112	0.21687	55.187	77.156	5.4091	0.77127	1.1239	174.82	27.098	5.4357e-06	0.0075045	vapor

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So, $\rho_g 1$ = 4.611 kg/m^3.= density of sat. vap.for Nitrogen at 0.1013 MPa (= 101.3 kPa).

By a similar procedure, for a pressure of 0.45 0 MPa (= 450 kPa), we get:

Data on Saturation Curve

Temperature (K)	Pressure (MPa)	Density (kg/m3)	Volume (m3/kg)	Internal Energy (kJ/kg)	Enthalpy (kJ/kg)	Entropy (J/g*K)	Cv (J/g*K)	Cp (J/g*K)	Sound Spd. (m/s)	Joule-Thomson (K/MPa)	Viscosity (Pa*s)	Therm. Cond. (W/m*K)	Surf. Tension (N/m)	Phase
92.670	0.45000	730.97	0.0013681	-90.332	-89.716	3.2095	1.0085	2.1758	689.67	-0.18832	9.4205e-05	0.11463	0.0055261	liquid

Vapor Phase Data

Data on Saturation Curve

Temperature (K)	Pressure (MPa)	Density (kg/m3)	Volume (m3/kg)	Internal Energy (kJ/kg)	Enthalpy (kJ/kg)	Entropy (J/g*K)	Cv (J/g*K)	Cp (J/g*K)	Sound Spd. (m/s)	Joule-Thomson (K/MPa)	Viscosity (Pa*s)	Therm. Cond. (W/m*K)	Phase
92.670	0.45000	18.647	0.053627	61.934	86.066	5.1064	0.81846	1.3133	182.54	19.845	6.6935e-06	0.0094357	vapor

i.e. $\rho_g 2$ = 18.647 kg/m^3 = density of sat. vap. for Nitrogen at 450 kPa

Therefore:

$$\Delta m_g := V2 \cdot \rho_{g2} - V1 \cdot \rho_{g1} \quad \text{...kg ...}$$

i.e. $\Delta m_g = 1.197 \cdot 10^3$ **kg.... amount of pressurization gas required....Ans.**

==========

Note: *Other three good references* for properties of cryogenic fluids are Ref. [5], [6], and [7]. References [5] and [6] give not only properties of fluids, but also their applications, and several calculators to convert liquid and gas properties. Ref. [7] gives properties of Air, N2, O2, CO2, NH3, CH4, C2H6, C2H2F4. It also has an almost complete steel index.

Prob.6.12. A safety valve for nitrogen service has a free area of 30 cm^2 and a discharge coefficient of 0.650. The nitrogen (specific-heat ratio = 1.40) enters the valve at 250 K (450°R) and 450 kPa absolute (65.3 psia). Determine the capacity of the valve.[1]

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Solution

Required size of the safety valve is determined from:

$$A_v = \frac{m_g \cdot \left(\frac{R_u \cdot T}{M}\right)^{\frac{1}{2}}}{C \cdot K_D \cdot p_{max}}$$

where:

$A_v := 30 \cdot 10^{-4}$ m^2 ... discharge area of valve

m_g = "mass flow rate through valve....to be determined"

$R_u := 8.314$ kg/kmol.K .... universal gas const.

$T := 250$ K .... abs. temp of gas

$K_D := 0.65$ ...discharge coeff. of valve, from test

$M := 28.013$ kg/kmol....for Nitrogen gas

p_{max} = (set gauge pressure * 1.1) + atm. pressure

i.e. $p_{max} := (450 - 101.3) \cdot 1.1 + 101.3$ kPa

i.e. $p_{max} = 484.87$ kPa

$\gamma := 1.4$ ...sp. heat ratio for N2

$$C := \left[\gamma \cdot \left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma+1}{\gamma-1}}\right]^{\frac{1}{2}}$$ i.e. $C = 0.685$

Therefore:

$$m_g := \frac{A_v \cdot C \cdot K_D \cdot p_{max}}{\left(\frac{R_u \cdot T}{M}\right)^{\frac{1}{2}}}$$

i.e. $m_g = 0.07516$ **kg/s ....Ans.**

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6.3 INSULATIONS FOR THE CRYOGENIC DEWAR:[1, 9, 10, 11]

Different types of insulations used are: (1) expanded foams, (2) gas-filled powders and fibrous materials, (3) vacuum alone, (4) evacuated powders and fibrous materials, (5) opacified powders, and (6) multilayer insulations. These insulations are listed in order of increasing performance and, generally, in order of increasing cost.

6.3.1 EXPANDED FOAM INSULATION [1]

It is cellular in structure. Thermal conductivity depends on the foaming gas used in manufacture. Generally, CO2 is the gas used. At Liquid nitrogen temperatures, CO2 will condense and with time, air diffuses in to the cells and the thermal conductivity may increase by as much as 40%. Further, large thermal contraction of the foam insulation is another serious disadvantage.

Apparent thermal conductivities of some foam insulations between 300 K and 77 K are given below [1]:

Table 6.10 Apparent thermal conductivity of foam insulations for boundary temperatures of 300 K (80°F) and 77 K (−139°F)

	Density		Thermal Conductivity	
Foam	kg/m^3	lb_m/ft^3	mW/m-K	Btu/hr-ft-°F
Polyurethane	11	0.70	33	0.019
Polystyrene	39	2.4	33	0.019
	46	2.9	26	0.015
Rubber	80	5.0	36	0.021
Silica	160	10.0	55	0.032
Glass	140	8.7	35	0.020

6.3.2 GAS FILLED POWDERS AND FIBROUS INSULATIONS [1]

Here, convection is avoided due to the small size of voids within the material. But, one disadvantage of this type is the possibility of migration of moisture and air to the lower temperature side, unless a vapour barrier is used.

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Some thermal conductivity values for fibre glass, powdered cork, perlite (a silica powder), Santocel, rock wool, and Vermiculite are given below:

Table 6.11 Apparent thermal conductivity of gas-filled powders and fibrous insulations for boundary temperatures of 300 K (80°F) and 90 K (−298°F)

	Density		Thermal Conductivity	
Insulation	kg/m^3	lb_m/ft^3	mW/m-K	Btu/hr-ft-°F
Perlite	50	3.1	26	0.015
	210	13.1	44	0.025
Silica aerogel	80	5.0	19	0.011
Vermiculite	120	7.5	52	0.030
Fiber glass	110	6.9	25	0.014
Rock wool	160	10.0	35	0.020

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6.3.3 VACUUM INSULATION [1]

Vacuum insulation is used extensively for small laboratory-size dewars. The use of vacuum insulation eliminates solid conduction and gaseous convection heat transfer. Heat is transferred across the annular space of a vacuum-insulated vessel by radiation from the hot outer jacket to the cold inner vessel and by gaseous conduction through the residual gas within the annular space.

Radiant heat transfer between two surfaces is given by modified Stefan-Boltzmann equation:

$$Q = F_e \cdot F_{12} \cdot \sigma \cdot A_1 \cdot \left(T_2^4 - T_1^4 \right)$$

where:

F_e = emissivity factor

F_{12} = configuration factor

σ = Stefan - Boltzmann constant = $5.669 * 10^{-8}$ W/m^2.K^4

A_1 = area of surface 1

T = absolute temp.

For cryogenic storage vessels, when inner surface 1 is completely enclosed by the outer surface 2, the configuration factor $F_{1\text{-}2}$ is equal to 1.

And **emissivity factor F_e for concentric spheres and cylinders is given by:**

$$\frac{1}{F_e} = \frac{1}{e_1} + \frac{A_1}{A_2} \cdot \left(\frac{1}{e_2} - 1 \right)$$

where e is the emissivity of the surface, and A is the surface area.

Radiant heat transfer can be reduced by interposing floating **radiation shields** of highly reflecting surfaces in between the hot and cold surfaces. For parallel plates, if N number of shields of emissivity e_s are placed between the hot and cold surfaces, the emissivity factor is given by:

$$\frac{1}{F_e} = \left(\frac{1}{e_1} + \frac{1}{e_s} - 1 \right) + (N-1) \cdot \left(\frac{2}{e_s} - 1 \right) + \left(\frac{1}{e_2} + \frac{1}{e_s} - 1 \right)$$

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Heat transfer by 'gaseous conduction':

In a vacuum space, in addition to heat transfer by radiation, conduction by residual gas molecules has also to be considered, if the mean free path of the molecules is larger than the vacuum space (i.e. distance between inner and outer vessels) in a dewar. In this 'free molecular conduction', molecules do not strike each other, but individually transfer energy from one surface to the other. So, an **'accommodation coefficient (a)'** is defined as follows:

a = (actual energy transfer) / (max. possible energy transfer)

Approximate accommodation coefficients for a given gas and temperatures are given below:

Table 6.12 Approximate accommodation coefficients

Temperature		Gas			
K	°R	Helium	Hydrogen	Neon	Air
300	540	0.29	0.29	0.66	0.8–0.9
78	140	0.42	0.53	0.83	1.00
20	36	0.59	0.97	1.00	1.00

By permission from R. B. Scott, *Cryogenic Engineering*, D. Van Nostrand Co., Princeton, N.J., 1959.

The **accommodation coefficient factor F_a** between inner and outer surfaces is defined by

$$\frac{1}{F_a}=\frac{1}{a_1}+\left(\frac{A_1}{A_2}\right)\cdot\left(\frac{1}{a_2}-1\right)$$

where: subscript 1 denotes the enclosed surface and the subscript 2 denotes the enclosure.

And, the energy transfer rate by molecular conduction is determined from:

$$Q=G\cdot P\cdot A_1\cdot\left(T_2-T_1\right)$$

where:

$$G=\frac{\gamma+1}{\gamma-1}\cdot\left(\frac{R}{8\cdot\pi\cdot T}\right)^{\frac{1}{2}}\cdot F_a$$

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where:

R = specific gas constant = R_u / M

Ru = universal gas constant = 8.31434 J/mol.K

M = Mol. wt. of gas

γ = sp. heat ratio = cp / cv

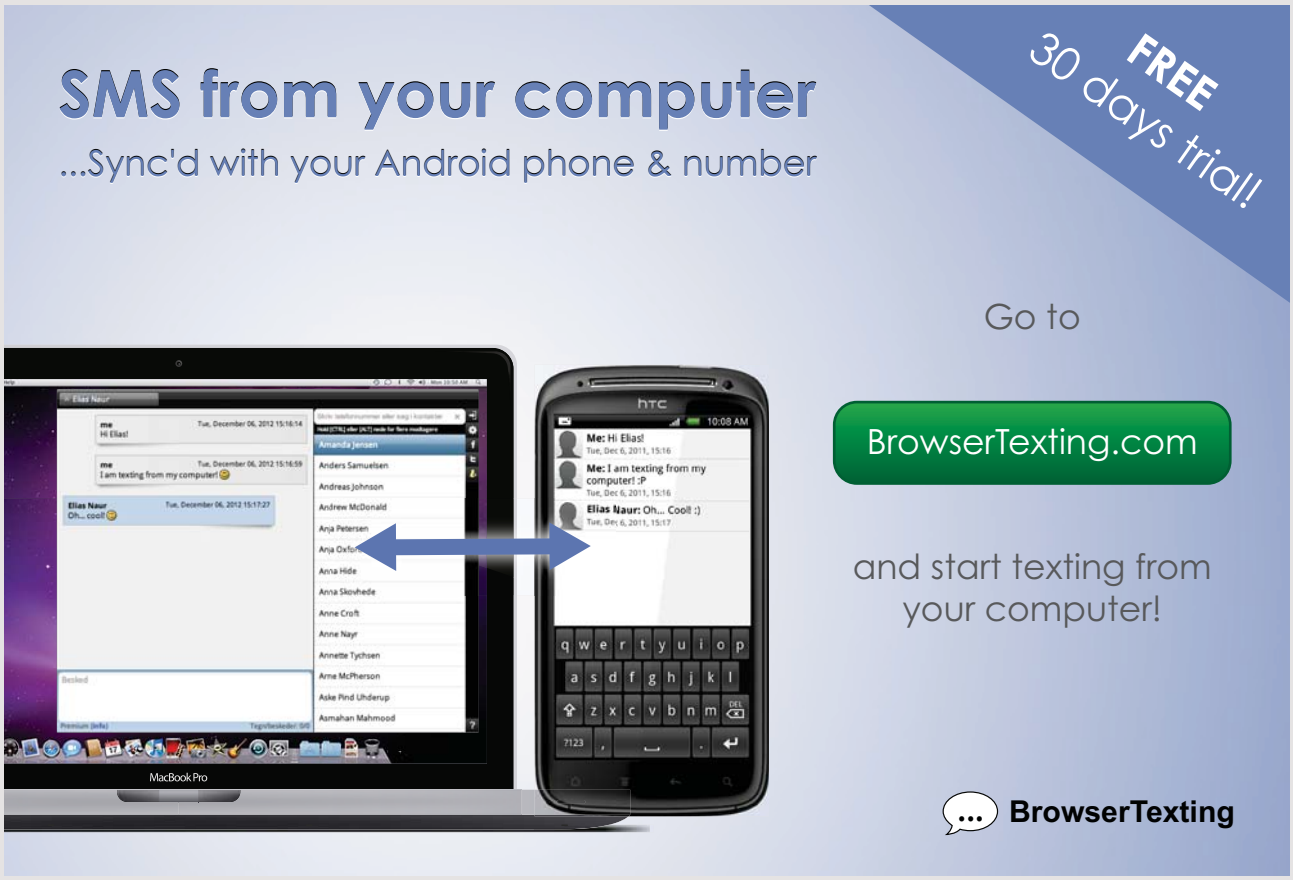
Remember: for free molecular conduction to occur, the mean free path of the gas molecules should be larger than the spacing between the two surfaces. To check this, mean free path is determined from:

$$\lambda = \left(\frac{\mu}{p}\right)\left(\frac{\pi \cdot R \cdot T}{2}\right)^{\frac{1}{2}}$$

where:

μ = gas viscosity at temperature, T

p = absolute pressure of gas.



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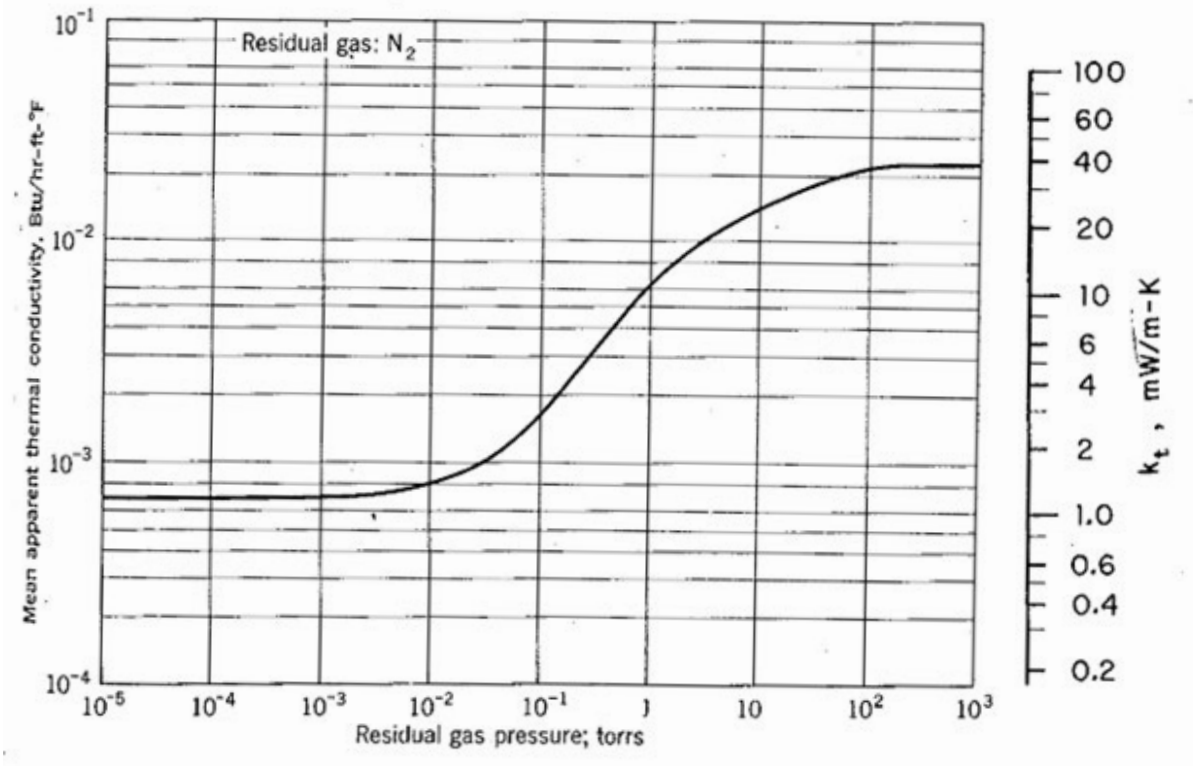
Since gaseous conduction heat transfer is directly proportional to the pressure, its value can be made negligible compared to radiant heat transfer, by having sufficiently low values of pressure in the space between the inner and outer vessels.

6.3.4 EVACUATED POWDER AND FIBROUS INSULATION [1]

Evacuating the gas from the powder and fibrous insulation is an obvious method to reduce the heat transfer rate in these insulations.

Following fig shows the variation of mean apparent thermal conductivity for 30 to 80 mesh perlite as the residual gas pressure is reduced:

Fig. 6.4 Variation of mean apparent thermal conductivity with residual gas pressure for an evacuated-powder insulation (30 to 80 mesh perlite). The residual gas is nitrogen, and the boundary temperatures are 300 K (540°R) and 78 K (140°R).



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Thermal conductivity values for several evacuated porous insulations are given in the following Table:

Table 6.13 Thermal conductivity for evacuated powders and fibrous materials. The boundary temperatures are 300 K (80°F) and 77.4 K (−321°F), and the residual gas pressure is below 0.13 Pa (0.001 torr).

	Density		Thermal conductivity	
Insulation	kg/m³	lb_m/ft^3	mW/m-K	Btu/hr-ft-°F
Fine perlite	180	11.2	0.95	0.55×10^{-3}
Coarse perlite	64	4.0	1.90	1.10×10^{-3}
Silica aerogel	80	5.0	1.60	0.92×10^{-3}
Calcium silicate	210	13.1	0.59	0.34×10^{-3}
Lampblack	200	12.5	1.20	0.69×10^{-3}
Fiberglass	50	3.1	1.70	0.98×10^{-3}

Heat transfer rate through evacuated powders and fibrous insulations is determined from:

$$Q = k_m \cdot A_m \cdot \frac{(T_h - T_c)}{\Delta x}$$

where:

Δx = thickness of insulation,

T_h, T_c = temperatures of warm and cold surfaces,

A_m = mean heat transfer area of insulation

And,

A_m is given by:

$A_m = (A_2 - A_1) / \ln(A_2/A_1)$ .... **for concentric cylinders, and**

$A_m = (A_1 * A_2)^{(1/2)}$ ...**for insulation between heads, and concentric spheres**

where A_1 is the inner (i.e. enclosed) surface area and A_2 is the outer (i.e. enclosure) surface area.

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6.3.5 OPACIFIED POWDER INSULATION [1]

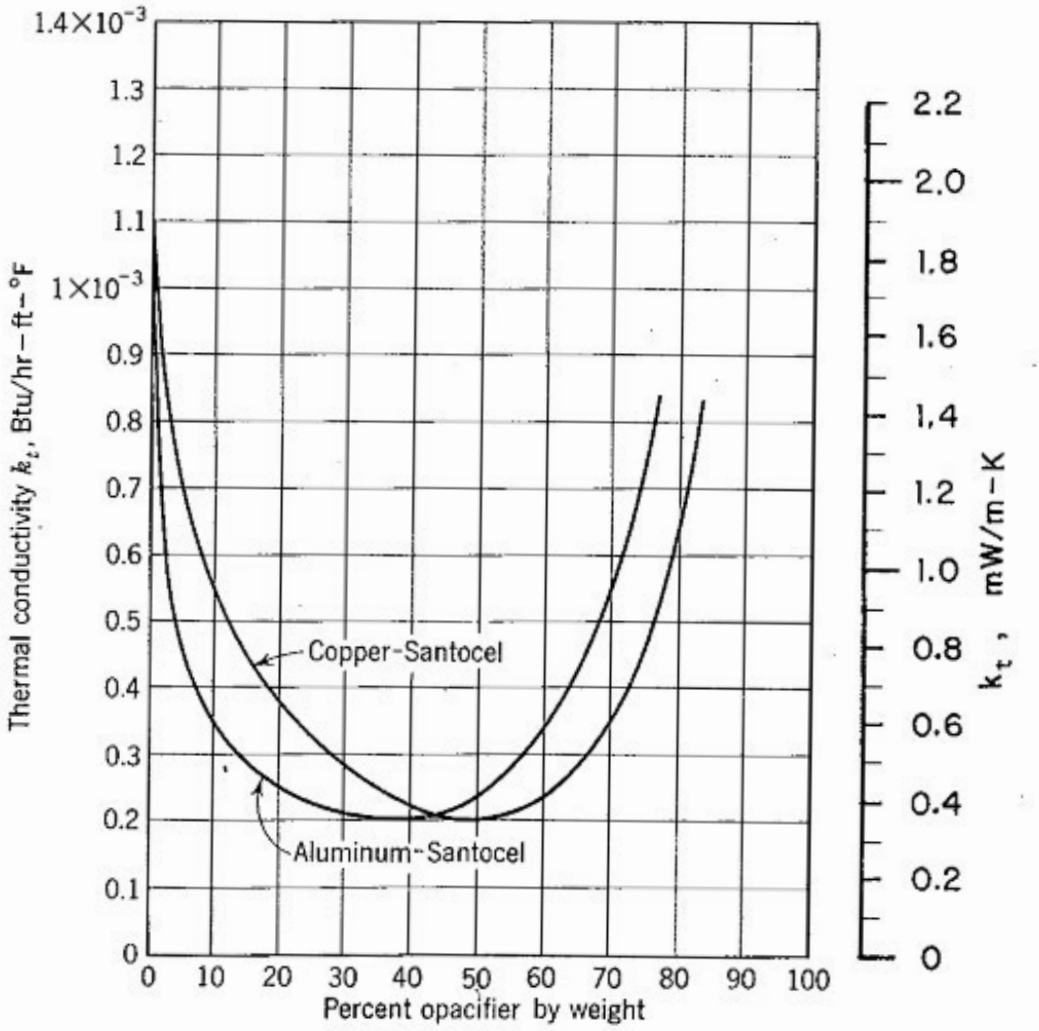
Since the major mode of heat transfer in an evacuated powder is by radiation, efforts are made to reduce this by adding copper or aluminium flakes in to the evacuated powder.

Apparent thermal conductivity of two opacified powders is shown below:

From a safety point of view, copper flakes are preferable compared to aluminium flakes, particularly for oxygen containers.

Opacified powders have the *disadvantage* that vibration can cause the metal flakes to pack together and result in a 'thermal short' and increase the thermal conductivity.

Fig. 6.5 Variation of thermal conductivity with percent opacifier for opacified-powder insulations (Riede and Wang 1960).



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Thermal conductivities of a few opacified powder insulations are shown below:

Table 6.14 Thermal conductivity of opacified powder insulations. The boundary temperatures are 300 K (80°F) and 77.4 K (−321°F), and the residual gas pressure is below 0.13 Pa (0.001 torr).

	Density		Thermal Conductivity	
Insulation	kg/m^3	lb$_m$/ft^3	mW/m-K	Btu/hr-ft-°F
50/50 w/w Copper-Santocel	180	11.2	0.33	0.19×10^{-3}
40/60 w/w Aluminum-Santocel	160	10.0	0.35	0.20×10^{-3}
50/50 w/w Bronze-Santocel	179	11.2	0.58	0.33×10^{-3}
Silca-carbon	80	5.0	0.48	0.28×10^{-3}

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6.3.6 MULTILAYER INSULATIONS [1]

Multilayer insulations consist of alternating layers of a highly reflecting material, such as aluminum foil, copper foil, or aluminized Mylar, and a low-conductivity spacer, such as fiber-glass mat or paper, glass fabric, or nylon net. Multilayer insulations must be evacuated to pressures below 10 mPa (7.5 x 10^{-5} torr) to be eflective. The dependence to the apparent thermal conductivity on residual gas pressure for a typical multilayer insulation is shown in Fig. 6.6.

And, thermal conductivity values for a few multilayer insulations are given in Table 6.15.

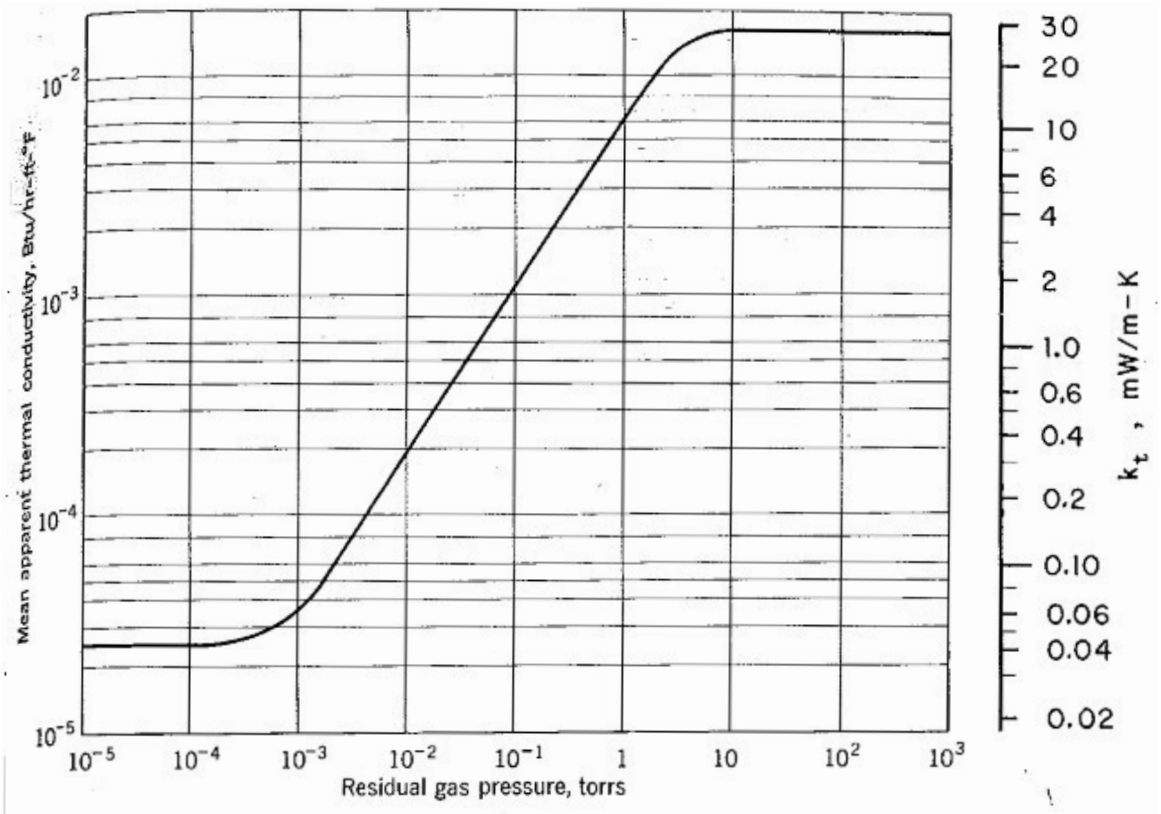

Fig. 6.6 Variation of mean apparent thermal conductivity with residual gas pressure for a typical multilayer insulation. The insulation layer-density is 24 layers/cm (60 layers/in.), and the boundary temperatures are 300 K (540°R) and 90.5 K (163°R).

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Table 6.15 Thermal conductivity for multilayer insulations for boundary temperatures of 300 K (80°F) and 77.4 K (−321°F) with residual gas pressures of 1.3 mPa (10^{-5} torr)

Insulation	Layer Density		Thermal Conductivity	
	layer/cm	layer/in.	μW/m-K	Btu/hr-ft-°F
0.006-mm aluminum foil + 0.15-mm Fiberglass paper	20	50	37	2.1×10^{-5}
0.006-mm aluminum foil + 2-mm mesh rayon net	10	25	78	4.5×10^{-5}
0.006-mm aluminum foil + 2-mm mesh nylon net	11	28	34	2.0×10^{-5}
NRC-2 crinkled aluminized Mylar film 0.006 mm	35	89	42	2.4×10^{-5}
Dimplar dimpled + smooth Mylar film	8	20	42	2.4×10^{-5}
0.0087-mm aluminum foil + carbon-loaded glass-fiber paper[a]	30	76	14	0.85×10^{-5}

[a]Residual gas pressure = 0.4 mPa = 3×10^{-6} torr

Last item in Table 6.15 above needs special mention: It has a very low thermal conductivity of about 14 micro-Watt/m.K, and was developed by **Prof. Scurlock** (*this author's teacher and guide*) **and Saull in 1976**. Since the out-gassing of the shields in a multilayer insulation can cause deterioration of vacuum, they devised an ingenious method of using carbon-filled glass-fiber paper for the spacer material. The carbon acted as a getter to adsorb the out-gassing load and maintain a low pressure within the insulation.

Typical densities of multilayer insulations range from 32 kg/m^3 to 320 kg/m^3 with layer densities ranging from 8 layers/cm to 40 layers/cm.

Variation of apparent thermal conductivity with layer density for a typical multilayer insulation is shown in Fig.6.7.

For a well evacuated multilayer insulation, apparent thermal conductivity may be calculated from the following equation:

$$k_t = \left(\frac{N}{\Delta x}\right)^{-1} \left[h_c + \frac{\sigma \cdot e \cdot \left(T_h^2 + T_c^2\right) \cdot \left(T_h + T_c\right)}{2 - e} \right]$$

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where:

h_c = solid conductance for spacer material

σ = Stefan - Boltzmann constant

e = effective emissivity of shield material

T_h, T_c = boundary temperatures of the insulation

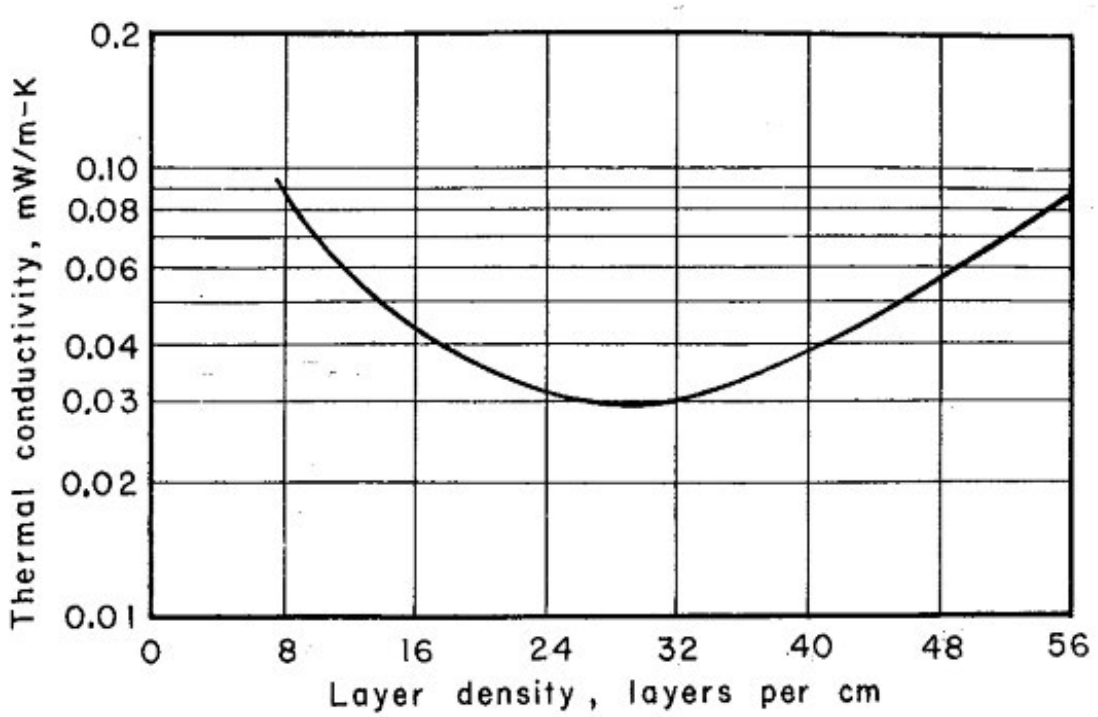

Fig. 6.7 Variation of thermal conductivity with layer density for a typical multilayer insulation (Wang 1961). The boundary temperatures are 294 K (530°R) and 78 K (140°R).

Thermal conductivity of multilayer insulations in the direction parallel to the shields is about 3 orders of magnitude greater than that in a direction normal to the shields. Therefore, extreme care must be exercised while applying the insulation, particularly around penetrations through the insulation.

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6.3.7 COMPARISON OF INSULATIONS [1]

A comparison of the *advantages and disadvantages* of the insulations used in cryogenic systems is given in the following summary.

	Advantages	Disadvantages
1	*Expanded foams*: Low cost. No need for rigid vacuum jacket. Good mechanical strength.	High thermal contraction. Conductivity may change with time.
2	*Gas-filled powders and fibrous materials*: Low cost. Easily applied 10 irregular shapes. Not flammable.	Vapor barrier is required. Powder can pack and conductivity is increased.
3	*Vacuum alone*: Complicated shapes may be easily insulated. Small cool-down loss. Low heat flux for small thickness between inner and outer vessel.	A permanent high vacuum is required. Low-emissivity boundary surfaces needed.
4	*Evacuated powders and fibrous materials*: Vacuum level less stringent than for multilayer insulations. Complicated shapes may be easily insulated. Relatively easy to evacuate.	May pack under vibratory loads and thermal cycling. Vacuum filters are required. Must be protected when exposed to moist air (retains moisture).
5	*Opacified powders :* Better performance than straight evacuated powders. Complicated shapes may be easily insulated. Vacuum requirement is not as stringent as for multilayer insulations and vacuum alone.	Higher cost than evacuated powders. Explosion hazards with aluminum in an oxygen atmosphere. Problems of settling of metallic flakes.
6	*Multilayer insulations:* Best performance of all insulations. Low weight. Lower cool-down loss compared with powders. Better stability than powders.	High cost per unit volume. Difficult to apply to complicated shapes. Problems with lateral conduction. More stringent vacuum requirements than powders.

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6.4 VAPOR-SHIELDED VESSELS [1]

Another method of reducing the heat in-leak to a cryogenic fluid storage vessel is to use the cold vent gas to refrigerate an intermediate shield, as shown in Fig. 6.8. The escaping vent gas intercepts some of the heat that would otherwise find its way to the product liquid. The effectiveness of this method in reducing the heat in-leak depends upon the ratio of sensible heat absorbed by the vent gas to the latent heat of the fluid.

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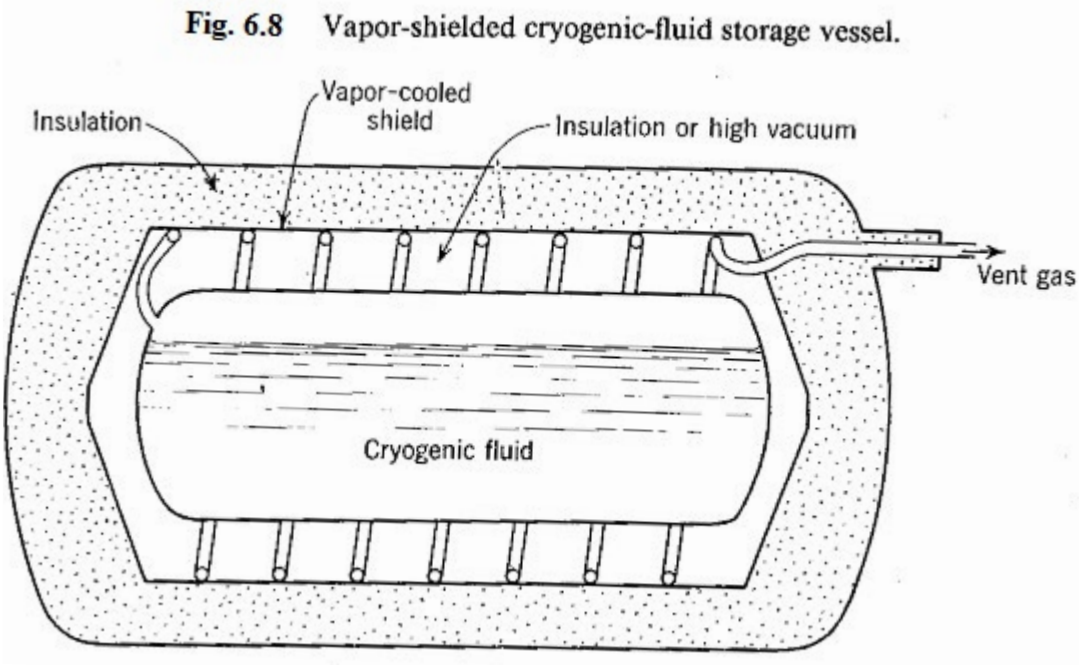

Fig. 6.8 Vapor-shielded cryogenic-fluid storage vessel.

Let:

T_1 = Temperature of the inner vessel

T_2 = Temperature of outer vessel

T_s = Temperature of the shield

Then, dimensionless temp of the shield is given by:

$$\theta = \frac{\pi 2 + 1}{2 \cdot \pi 1 \cdot \pi 2} \cdot \left[\left[1 + \frac{4 \cdot \pi 1 \cdot \pi 2}{(\pi 2 + 1)^2} \right]^{\frac{1}{2}} - 1 \right]$$

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where:

$$\theta = \frac{T_s - T_1}{T_2 - T_1}$$

$$\pi 1 = \frac{c_p \cdot (T_2 - T_1)}{h_{fg}}$$

$$\pi 2 = \frac{U_1}{U_2}$$

where U_2 = conductance between the outer surface and the shield, given by:

$Q_{2\text{-}s} = U_2 \cdot (T_2 - T_s)$

U_1 = conductance between the shield and the inner surface, given by:

$Q_{s\text{-}1} = U_1 \cdot (T_s - T_1) = m_g \cdot h_{fg}$ where m_g = boil off rate, h_{fg} = heat of vap. of fluid

Further: the conductance U_2 between the outer surface and the shield is given by:

$$U_2 = \left(\frac{k_t \cdot A}{\Delta x}\right)_{ins} + \left(\frac{k_t \cdot A}{\Delta x}\right)_{sup} + \left(\frac{k_t \cdot A}{\Delta x}\right)_{piping}$$

where: k_t = thermal cond. of insulation, supports or piping;

A = heat transfer area for each of these components, and

Δx = length of conduction path, i.e. thickness of insulation, support length for vessel supports, and piping length

Variation of shield temp with $\pi 2$ for selected values of parameter $\pi 1$ is shown in Fig.6.9.

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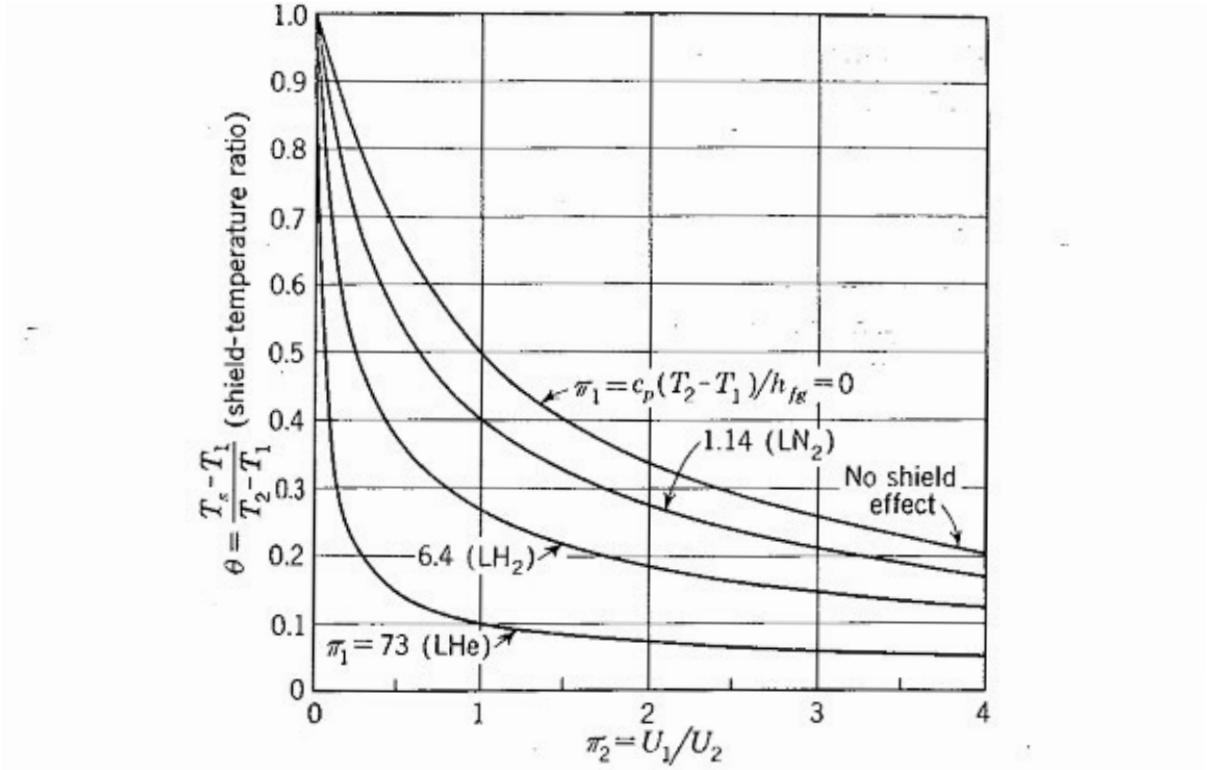

Fig. 6.9 Variation of the shield temperature ratio with conductance ratio for a vapor-shielded cryogenic-fluid storage vessel.

Values of $\pi 1$ for an ambient temp of 21deg. C are shown below:

Fluid	$\pi_1 = c_p(T_2 - T_1)/h_{fg}$
Helium	72.9
Hydrogen	6.37
Neon	6.40
Nitrogen	1.14
Oxygen	0.87

Note that vapor shielding is most effective for liquid helium vessels, and not so effective for other fluids.

If the mass flow rate through the shield were zero, then shield temp would be:

$$\theta_0 = \frac{1}{\pi 2 + 1}$$

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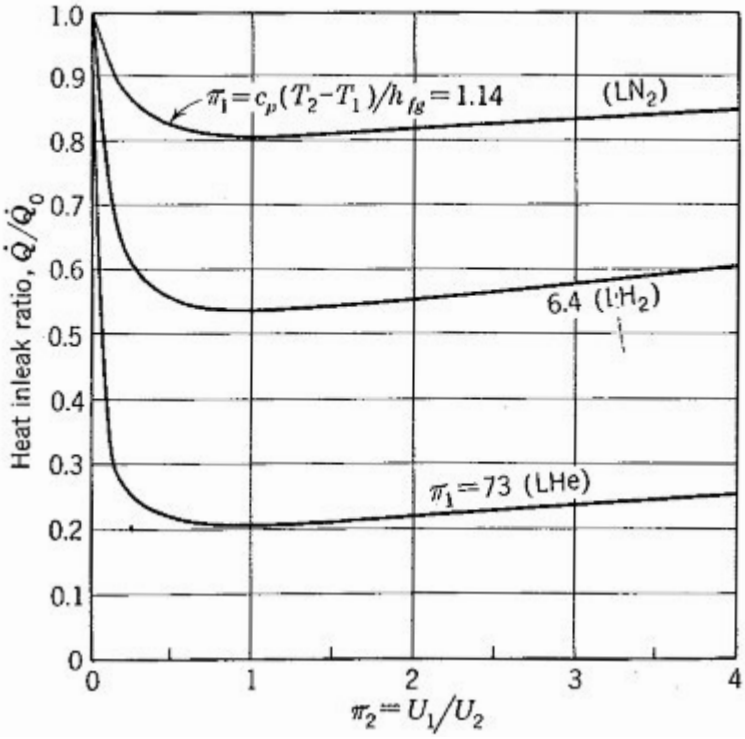
The ratio of heat in-leak with vapor shielding to heat in-leak without vapor shielding is given by:

$$\frac{Q}{Q_o} = \frac{\theta}{\theta_o}$$

This expression for Q/Q_o (= θ/θ_o) is plotted in Fig. 6.10 for various values of $\pi 1$:

Fig. 6.10 Variation of the heat-inleak ratio (heat inleak with vapor shielding/heat inleak with no shield) with conductance ratio for a vapor-shielded cryogenic-fluid storage vessel.



Note from the above Fig. that for liquid helium (i.e. $\pi 1$ = 73), Q/Q_0 is 0.2 to 0.25, i.e. by using a vapor shield, the heat in-leak is reduced by a factor of 4 to 5, thus saving the boil off fluid. For liquid nitrogen, this factor is about 1.2.So, vapor shield may not be worth the extra trouble and expense since LN2 is relatively cheap compared to LHe.

Prob.6.13. A cylindrical liquid-nitrogen vessel is insulated with perlite (k_t = 1.50mW/m-K -= 8.67 x 10^{-4} Btu/hr-ft-°F). The inner-vessel outside diameter is 1.842m (72.5 in.), and the outer-vessel inside diameter is 2.450 m (96.5 in.). Both vessels have 2:1 elliptical heads. The cylindrical portions of the vessel have lengths as follows: inner vessel, 24 m (78.74 ft); outer vessel, 24.8 m (81.36 ft). The outer vessel temperature is 300 K (540°R), and the inner-vessel temperature is 77.8 K (140°R). Determine the heat-transfer rate through the insulation.[1]

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Solution:

For the cylindrical vessel:

$D_1 := 1.842$ m ....OD of inner vessel

$D_2 := 2.45$ m .... IDof outer vessel

$L_1 := 24$ m .... cyl length of inner vessel

$L_2 := 24.8$ m ... cyl length of outer vessel

$T_1 := 77.8$ K .... temp of inner vessel

$T_2 := 300$ K ... temp of outer vessel

Elliptical heads, 2:1 for both inner and outer vessels

$k_t := 1.5 \cdot 10^{-3}$ W/m.K ...thermal cond. of perlite insulation

Note: for elliptical heads, outside surface area of the head = 0.345. π . D_o^2 (from Table 6.3)

And, dish height is given by:

Ref: https://www.slideshare.net/ajaytyouharia/cedppt

DH = D_i / 4 (see the fig below):

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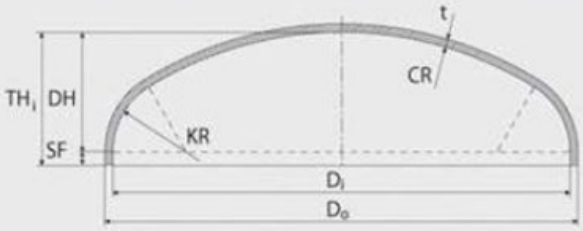

Shape 2:1	Shape 1,9:1
$D_i = D_o - 2 \times t$	$D_i = D_o - 2 \times t$
$CR = 0{,}9 \times D_i$	$CR = D_i / 1{,}16$
$KR = 0{,}17 \times D_i$	$KR = D_i / 5{,}39$
SF = acc. specification	SF = acc. NF E 81-103
$DH = 0{,}25 \times D_i$	$DH = D_i / 3{,}8$
$TH_i = SF + DH$	$TH_i = SF + DH$

Where:

t = wall thickness, D_o = outside diameter, D_i = inside diameter, SF = straight flange height, DH = dished height, TH_i = total height inside, CR = crown radius, KR = knuckle radius

Fig. 6.11 Elliptical head-- dimensions

Therefore:

$A_1 := \pi \cdot D_1 \cdot L_1$ m^2 .... surface area of inner vessel

i.e. $A_1 = 138.884$ m^2

$A_2 := \pi \cdot D_2 \cdot L_2$ m^2 .... surface area of outer vessel

i.e. $A_2 = 190.883$ m^2

Therefore, mean area of insulation around the inner shell is:

$$A_m := \frac{A_2 - A_1}{\ln\left(\frac{A_2}{A_1}\right)}$$ i.e. $A_m = 163.508$ m^2

Thickness of insulation around the shell is:

$$\Delta x := \frac{D_2 - D_1}{2}$$ i.e. $\Delta x = 0.304$ m

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Therefore, heat transfer rate through insulation of the cylindrical portion is:

$$Q_{cyl} := k_t \cdot A_m \cdot \frac{(T_2 - T_1)}{\Delta x}$$

i.e. $Q_{cyl} = 179.267$ **W... for cyl. part**

Now, for the Elliptical (2:1) heads, total area of two inner cyl heads:

$$A_{innerheads} := 2 \cdot (0.345 \cdot \pi \cdot D_1^2)$$

i.e. $A_{innerheads} = 7.355$ m^2

Similarly, total area of two outer heads:

$$A_{outerheads} := 2 \cdot (0.345 \cdot \pi \cdot D_2^2)$$

i.e. $A_{outerheads} = 13.012$ m^2

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And, the mean surface area for insulation between the heads is:

$$A_{mean} := \left(A_{innerheads} \cdot A_{outerheads}\right)^{\frac{1}{2}}$$

i.e. $A_{mean} = 9.783$ m^2

Thickness of insulation around the heads is:

$$\Delta x_{insheads} := \frac{D_2 - D_1}{4}$$ ....since dish height for 2:1 elliptical heads = D/4

i.e. $\Delta x_{insheads} = 0.152$ m

Therefore, heat transfer rate through insulation around the heads is:

$$Q_{heads} := k_t \cdot A_{mean} \cdot \frac{\left(T_2 - T_1\right)}{\Delta x_{insheads}}$$

i.e. $Q_{heads} = 21.451$ **W.. for two elliptical heads**

Therefore, total heat transfer rate through insulation:

$$Q_{tot} := Q_{cyl} + Q_{heads}$$

i.e. $Q_{tot} = 200.718$ **W ... total heat transfer rate through insulation ... Ans.**

==========

Prob. 6.14. Write a Mathcad Function to calculate heat transfer through insulation only, for a cylindrical tank with elliptical heads.

Following is the Mathcad Function: (Inputs are already defined in the above problem).

$$Qins\_tot\_cyl\_ellipt\left(D_1, D_2, L_1, L_2, T_1, T_2, k_t\right) :=$$

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$$A_1 \leftarrow \pi \cdot D_1 \cdot L_1$$

$$A_2 \leftarrow \pi \cdot D_2 \cdot L_2$$

$$A_m \leftarrow \frac{A_2 - A_1}{\ln\left(\frac{A_2}{A_1}\right)}$$

$$\Delta x \leftarrow \frac{D_2 - D_1}{2}$$

$$Q_{cyl} \leftarrow k_t \cdot A_m \cdot \frac{(T_2 - T_1)}{\Delta x}$$

$$A_{innerheads} \leftarrow 2 \cdot \left(0.345 \cdot \pi \cdot D_1^{\,2}\right)$$

$$A_{outerheads} \leftarrow 2 \cdot \left(0.345 \cdot \pi \cdot D_2^{\,2}\right)$$

$$Am_{heads} \leftarrow \left(A_{innerheads} \cdot A_{outerheads}\right)^{\frac{1}{2}}$$

$$\Delta x_{insheads} \leftarrow \frac{D_2 - D_1}{4}$$

$$Q_{heads} \leftarrow k_t \cdot Am_{heads} \cdot \frac{(T_2 - T_1)}{\Delta x_{insheads}}$$

$$Q_{tot} \leftarrow Q_{cyl} + Q_{heads}$$

"Acyl1(m2)"	"Acyl2(m2)"	"Amcyl(m2)"	"Dx_cyl(m)"	"Ahead1(m2)"	"Ahead2(m2)"	"Am_head(m2)"	"Dx_heads(m)"	"Qcyl(W)"	"Qheads(W)"	"Qtot(W)"
A_1	A_2	A_m	Δx	$A_{innerheads}$	$A_{outerheads}$	Am_{heads}	$\Delta x_{insheads}$	Q_{cyl}	Q_{heads}	Q_{tot}

Then, using this Mathcad Function, for the above problem, we get:

$$Qins\_tot\_cyl\_ellipt\left(D_1, D_2, L_1, L_2, T_1, T_2, k_t\right) =$$

	0	1	2	3	4	5	6	7	8	9	10
0	"Acyl1(m2)"	"Acyl2(m2)"	"Amcyl(m2)"	"Dx_cyl(m)"	"Ahead1(m2)"	"Ahead2(m2)"	"Am_head(m2)"	"Dx_heads(m)"	"Qcyl(W)"	"Qheads(W)"	"Qtot(W)"
1	138.884	190.883	163.508	0.304	7.355	13.012	9.783	0.152	179.267	21.451	200.718

The results are the same as in the earlier problem, as they should be.

==========

Prob. 6.15. Write a Mathcad Function to calculate heat transfer through insulation only, for a cylindrical tank with hemispherical ends:

Following is the desired Mathcad Function:

$$Qins\_tot\_cyl\_hemisph\left(D_1, D_2, L_1, L_2, T_1, T_2, k_t\right) :=$$

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$$A_1 \leftarrow \pi \cdot D_1 \cdot L_1$$

$$A_2 \leftarrow \pi \cdot D_2 \cdot L_2$$

$$A_m \leftarrow \frac{A_2 - A_1}{\ln\left(\frac{A_2}{A_1}\right)}$$

$$\Delta x \leftarrow \frac{D_2 - D_1}{2}$$

$$Q_{cyl} \leftarrow k_t \cdot A_m \cdot \frac{(T_2 - T_1)}{\Delta x}$$

$$A_{innerheads} \leftarrow 2 \cdot \left(\frac{\pi \cdot D_1^2}{2}\right)$$

$$A_{outerheads} \leftarrow 2 \cdot \left(\frac{\pi \cdot D_2^2}{2}\right)$$

$$Am_{heads} \leftarrow \left(A_{innerheads} \cdot A_{outerheads}\right)^{\frac{1}{2}}$$

$$\Delta x_{insheads} \leftarrow \frac{D_2 - D_1}{2}$$

$$Q_{heads} \leftarrow k_t \cdot Am_{heads} \cdot \frac{(T_2 - T_1)}{\Delta x_{insheads}}$$

$$Q_{tot} \leftarrow Q_{cyl} + Q_{heads}$$

"Acyl1(m2)"	"Acyl2(m2)"	"Amcyl(m2)"	"Dx_cyl(m)"	"Ahead1(m2)"	"Ahead2(m2)"	"Am_head(m2)"	"Dx_heads(m)"	"Qcyl(W)"	"Qheads(W)"	"Qtot(W)"
A_1	A_2	A_m	Δx	$A_{innerheads}$	$A_{outerheads}$	Am_{heads}	$\Delta x_{insheads}$	Q_{cyl}	Q_{heads}	Q_{tot}

Prob.6.16. Determine the heat transfer through the insulation for a 106 m3 (28 000 gal U.S.) liquid-oxygen storage vessel with the following characteristics. The insulation is evacuated perlite with an apparent thermal conductivity of 1.20 mW/m-K (0.69 x 10^{-3} Btu/hr-ft-°F). The inner shell has an OD of 3.070 m and a length of 14.0 m, and the inner-vessel hemispherical heads have an OD of 3.062 m. The outer shell has an ID of 3.636 m and a length of 14.0 m, and the outer-vessel hemispherical heads have an ID of 3.648 m. The temperature of the outer vessel may be assumed to be 300 K, and the temperature of the inner vessel may be assumed to be 90.2 K.[1]

Solution:

Above problem is part of a worked-out problem in Ref.[1]. Shown here to illustrate the use of the above Mathcad function and compare it with the results in Ref.[1]

Note: The simple Mathcad Function above, does not take in to account the thickness of the plates. Still, the results match quite well with those obtained in Ref.[1].

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Let us write the data:

For the cylindrical vessel:

$D_1 := 3.07$ m ....OD of inner vessel

$D_2 := 3.636$ m .... ID of outer vessel

$L_1 := 14$ m .... cyl length of inner vessel

$L_2 := 14$ m ... cyl length of outer vessel

$T_1 := 90.2$ K .... temp of inner vessel

$T_2 := 300$ K ... temp of outer vessel

Hemispherical heads, for both inner and outer vessels

$k_t := 1.2 \cdot 10^{-3}$ W/m.K ...thermal cond. of perlite insulation

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Then, using the Mathcad Function, we get:

$$Qins\_tot\_cyl\_hemisph\left(D_1, D_2, L_1, L_2, T_1, T_2, k_t\right) =$$

	0	1	2	3	4	5	6	7	8	9	10
0	"Acyl1(m2)"	"Acyl2(m2)"	"Amcyl(m2)"	"Dx_cyl(m)"	"Ahead1(m2)"	"Ahead2(m2)"	"Am_head(m2)"	"Dx_heads(m)"	"Qcyl(W)"	"Qheads(W)"	"Qtot(W)"
1	135.026	159.92	147.122	0.283	29.609	41.533	35.068	0.283	130.881	31.197	162.078

i.e.

Heat transfer rate in insulation in cylindrical part = Qcyl = 130.881 W,

Heat transfer rate in insulation in hemispherical heads = Qheads = 31.197 W, and

Total heat transfer through insulation alone = Qcyl + Qheads = 162.078 W

Now, comparing to the above results from the Mathcad Function, to the results in Ref.[1], corresponding values are:

135.0, 159.9, 147.1, 0.283, 29.46, 41.81, 35.09, 0.293, 130.881, 31.197, 162.078

We see that results match very well, though we have ignored the thickness of shell in the simple Mathcad Function.

==========

Prob.6.17. Write a Mathcad Function to determine Heat transfer rate in a vacuum insulated dewar. Also, include the heat transfer by gas conduction.

Solution:

Note:

Following is the explanation about various variables in this Mathcad Function:

D1 = OD of inner vessel, m; D2 = ID of outer vessel, m

T1 = temp of inner vessel, K; T2 = temp of outer vessel, K

a1 = accommodation coeff for air at temp T1; a2 = accommodation coeff for air at temp T2

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e1 = emissivity of inner vessel; e2 = emissivity of outer vessel

p = pressure, Pa; R_{gas} = gas constant for the gas (i.e. air), J/kg.K

γ = cp/cv = sp.heat ratio for gas; μ_{gas} = viscosity of gas at temp T2, Pa-s

$$Qins\_vacuum\_sph(D_1, D_2, T_1, T_2, e_1, a_1, e_2, a_2, p, R_{gas}, \mu_{gas}, \gamma) :=$$

$$A_1 \leftarrow \pi \cdot D_1^2$$

$$F_{12} \leftarrow 1$$

$$A_2 \leftarrow \pi \cdot D_2^2$$

$$F_e \leftarrow \frac{1}{\frac{1}{e_1} + \frac{A_1}{A_2} \cdot \left(\frac{1}{e_2} - 1\right)}$$

$$\sigma \leftarrow 5.669 \cdot 10^{-8}$$

$$Q_{rad} \leftarrow F_e \cdot F_{12} \cdot \sigma \cdot A_1 \cdot \left(T_2^4 - T_1^4\right)$$

$$mfp \leftarrow \left(\frac{\mu_{gas}}{p}\right) \cdot \left(\frac{\pi \cdot R_{gas} \cdot T_2}{2}\right)^{\frac{1}{2}}$$

$$\Delta x \leftarrow \frac{D_2 - D_1}{2}$$

$$Q_{gas} \leftarrow 0 \quad \text{if} \quad mfp < \Delta x$$

if $mfp \geq \Delta x$

$$F_a \leftarrow \frac{1}{\frac{1}{a_1} + \frac{A_1}{A_2} \cdot \left(\frac{1}{a_2} - 1\right)}$$

$$G \leftarrow \frac{\gamma + 1}{\gamma - 1} \cdot \left(\frac{R_{gas}}{8 \cdot \pi \cdot T_2}\right)^{\frac{1}{2}} \cdot F_a$$

$$Q_{gas} \leftarrow G \cdot p \cdot A_1 \cdot \left(T_2 - T_1\right)$$

$$Q_{tot} \leftarrow Q_{rad} + Q_{gas}$$

"A1(m2)"	"A2(m2)"	"Fe"	"Qrad(W)"	"mfp(m)"	"Dx(m)"	"Fa"	"Qgas(W)"	"Qtot(W)"
A_1	A_2	F_e	Q_{rad}	mfp	Δx	F_a	Q_{gas}	Q_{tot}

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Now, let us solve a problem using the above Function.

This is a worked-out problem in Barron, Ref.[1]. Solved with this Mathcad Function, just to illustrate the use of this function:

Prob.6.18. Determine the total heat-transfer rate from the outer shell to the inner shell of a spherical dewar with a 1.50 m (59 in.) OD inner shell and a 2.10 m (83 in.) ID outer shell. The outer shell is at 300 K (540°R); it has an emissivity of 0.10 and an accommodation coefficient of 0.90. The inner shell is at 78 K (140°R); it has an emissivity of 0.05 and an accommodation cofficient of 1.00. The gas within the annular space is air at a pressure of 2 mPa (1.5 x 10^{-5} torr) measured at 300 K. The dewar is insulated by vacuum alone.[1]

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Solution:

Data:

$D_1 := 1.5$ m $D_2 := 2.1$ m $T_1 := 78$ K $T_2 := 300$ K

$a_1 := 1$ $e_1 := 0.05$ $a_2 := 0.9$ $e_2 := 0.1$

$p := 2 \cdot 10^{-3}$ Pa $R_{gas} := 287$ J/kg.K

$\mu_{gas} := 18.47 \cdot 10^{-6}$ Pa-s.... at 300 K $\gamma := 1.4$

Using the Mathcad Function written above, we get:

$Qins\_vacuum\_sph(D_1, D_2, T_1, T_2, e_1, a_1, e_2, a_2, p, R_{gas}, \mu_{gas}, \gamma) =$

"A1(m2)"	"A2(m2)"	"Fe"	"Qrad(W)"	"mfp(m)"	"Dx(m)"	"Fa"	"Qgas(W)"	"Qtot(W)"
7.069	13.854	0.041	131.384	3.396	0.3	0.946	3.477	134.861

Thus:

Heat transfer rate by radiation = Q_{rad} = 131.384 W

Heat transfer by molecular gas conduction = Q_{gas} = 3.477 W, and

Total heat transfer, $Q_{tot} = Q_{rad} + Q_{gas}$ = 134.861 W

Prob.6.19. A spherical liquid hydrogen vessel has an inner-vessel outside diameter of 1.220 m (48 in.) and an outer-vessel inside diameter of 1.675 m (66 in.). The vessel is insulated by vacuum alone. The emissivity of the inner vessel is 0.04, and the emissivity of the outer vessel is 0.09. The gas in the vacuum space is air (R = 287 J /kg-K; γ = 1.40) at a pressure of 15 mPa (1.1 x 10^{-4} torr). The outer vessel temperature is 295 K (531°R), and the inner-vessel temperature is 20.5 K (36.9°R). Determine: (a) the heat-transfer rate by radiation; (b) the heat-transfer rate by molecular conduction; and (c) the radiation heat-transfer rate if an intermediate floating shield having a 1.45-m (57-in.) diameter and an emissivity on both sides of 0.05 were used.[1]

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Data:

$D_1 := 1.22$ m ....OD of inner vessel

$D_2 := 1.675$ m.... IDof outer vessel

$T_1 := 20.5$ K..... temp of inner vessel

$T_2 := 295$ K.... temp of outer vessel

$a_1 := 1$ ...accommodation coeff for air at 20.5 K

$e_1 := 0.04$ ...emissivity of inner vessel

$a_2 := 0.9$ ...accommodation coeff for air at 300 K

$e_2 := 0.09$ ...emissivity of outer vessel

$p := 15 \cdot 10^{-3}$ Pa.....Pressure

$R_{gas} := 287$ J/kg.K ... Gas const for air

$\gamma := 1.4$ = cp/cv = ratio of sp. heats

Then, using the Mathcad Function written above, we get:

$Qins\_vacuum\_sph(D_1, D_2, T_1, T_2, e_1, a_1, e_2, a_2, p, R_{gas}, \mu_{gas}, \gamma) =$

"A1(m2)"	"A2(m2)"	"Fe"	"Qrad(W)"	"mfp(m)"	"Dx(m)"	"Fa"	"Qgas(W)"	"Qtot(W)"
4.676	8.814	0.033	66.114	0.445	0.228	0.944	21.463	87.577

Thus:

Heat transfer rate by radiation = Q_{rad} = 66.114 W

Heat transfer by molecular gas conduction = Q_{gas} = 21.463 W, and

Total heat transfer, $Q_{tot} = Q_{rad} + Q_{gas}$ = 87.577 W

If one floating shield is used:

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(Ref: Fundamentals of Heat & Mass Transfer, by M.Thirumaleshwar, Publ:Pearson Edu, 2006, ... page 797.)

$D_3 := 1.47$ .... dia of shield

$e_{s1} := 0.05$ $e_{s2} := 0.05$ ....emissivities of both sides of shield

$\sigma := 5.669 \cdot 10^{-8}$ W/m^2.K^4 ..... Stefan Boltzmann const.

Then, equation for heat transfer in a spherical system, with one shield, is:

$$Q_{12\_oneshield} = \frac{A_1 \cdot \sigma \cdot \left(T_2^4 - T_1^4\right)}{\frac{1}{e_1} + \frac{A_1}{A_2}\cdot\left(\frac{1}{e_1} - 1\right) + \frac{A_1}{A_3}\cdot\left(\frac{1}{e_{s1}} + \frac{1}{e_{s2}} - 1\right)}$$

where: A_1, A_2, A_3 ... are the surface areas of inner sphere, outer sphere and the radiation shields respectively.

And, $A_1 := 4 \cdot \pi \cdot \left(\frac{D_1}{2}\right)^2$ $A_2 := 4 \cdot \pi \cdot \left(\frac{D_2}{2}\right)^2$ $A_3 := 4 \cdot \pi \cdot \left(\frac{D_3}{2}\right)^2$

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Then, we get:

$$Q_{12\_oneshield} := \frac{A_1 \cdot \sigma \cdot \left(T_1^4 - T_2^4\right)}{\frac{1}{e_1} + \frac{A_1}{A_2}\cdot\left(\frac{1}{e_1} - 1\right) + \frac{A_1}{A_3}\cdot\left(\frac{1}{e_{s1}} + \frac{1}{e_{s2}} - 1\right)}$$

i.e. $Q_{12\_oneshield} = -31.078$ **W... Ans.**

Note: -ve sign indicates that heat flow is surface 2 to surface 1, i.e. from outer sphere to the inner sphere.

Also, note that by using one shield, the radiation heat transfer is reduced from 66.114 W to 31.078 W.

Prob.6.20. Determine the pressure to which hydrogen gas (R = 4124 J /kg-K; γ =1.41) must be reduced in order that the heat flux by molecular conduction through the gas be limited to 30 W/m2 (9.5 Btu/hr-ft). The two surfaces enclosing the gas may be considered as infinite planes at temperatures of 300 K (540°R) and 78 K (140°R). The pressure gauge is at 300 K.[1]

Solution:

For molecular conduction to occur, the mean free path of gas, i.e. mfp must be greater than the spacing between the plates, Δx:

So, consider 2 parallel plates, with areas A1 and A2, space between them being Δx, and let the pressure be such that mfp > Δx.

Let: $A_1 := 1$ $A_2 := 1$ m^2

$T_1 := 78$ K $T_2 := 300$ K

Accommodation coefficients for hydrogen at 300 K and 78 K: (From Table)

$a_1 := 0.29$ accommodation coeff for hydrogen at 300 K

$a_2 := 0.53$ accommodation coeff for hydrogen at 78 K

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And:

$R_{gas} := 4124$ J/kg.K

$\gamma := 1.41$ ..ratio of sp. heats

$Q_{gas} := 30$ W/m^2..... by data

Then, we have:

$$F_a := \frac{1}{\frac{1}{a_1} + \frac{A_1}{A_2}\cdot\left(\frac{1}{a_2} - 1\right)}$$ i.e. $F_a = 0.231$ .....accommodation coeff factor

$$G := \frac{\gamma + 1}{\gamma - 1}\cdot\left(\frac{R_{gas}}{8\cdot\pi\cdot T_2}\right)^{\frac{1}{2}}\cdot F_a$$ i.e. $G = 1.003$

Now:

$$Q_{gas} = G\cdot p\cdot A_1\cdot\left(T_2 - T_1\right)$$

Therefore: $$p := \frac{Q_{gas}}{G\cdot A_1\cdot\left(T_2 - T_1\right)}$$

i.e. $p = 0.135$ **Pa ..... Ans.**

i.e. Hydrogen gas pressure between two parallel plates, kept at 300 K and 78 K, must be reduced to 0.135 Pa for the heat flux by molecular conduction to be limited to 30 W/m^2.

Prob.6.21. Heat is transferred by molecular conduction across a vacuum space with one intermediate floating shield. The pressure of the residual gas (helium, R = 2077 J/kg-K; γ = 1.667) is 0.120 Pa (9 x 10-4 torr) as measured by a gauge at 300 K (540°R). The warm surface is at 300 K (540°R) and has an accommodation coefficient of 0.30; the cold surface is at 22 K (39.6°R) and has an accommodation coeflicient of 0.60. The shield material has an accommodation coefficient of 0.40.Determine the heat flux by molecular conduction, W/m2, assuming that the surfaces may be treated as infinite parallel plates. [1]

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Solution:

Consider 2 parallel plates, with areas A1 and A2, space between them being Δx, and let the pressure be such that mfp > Δx.

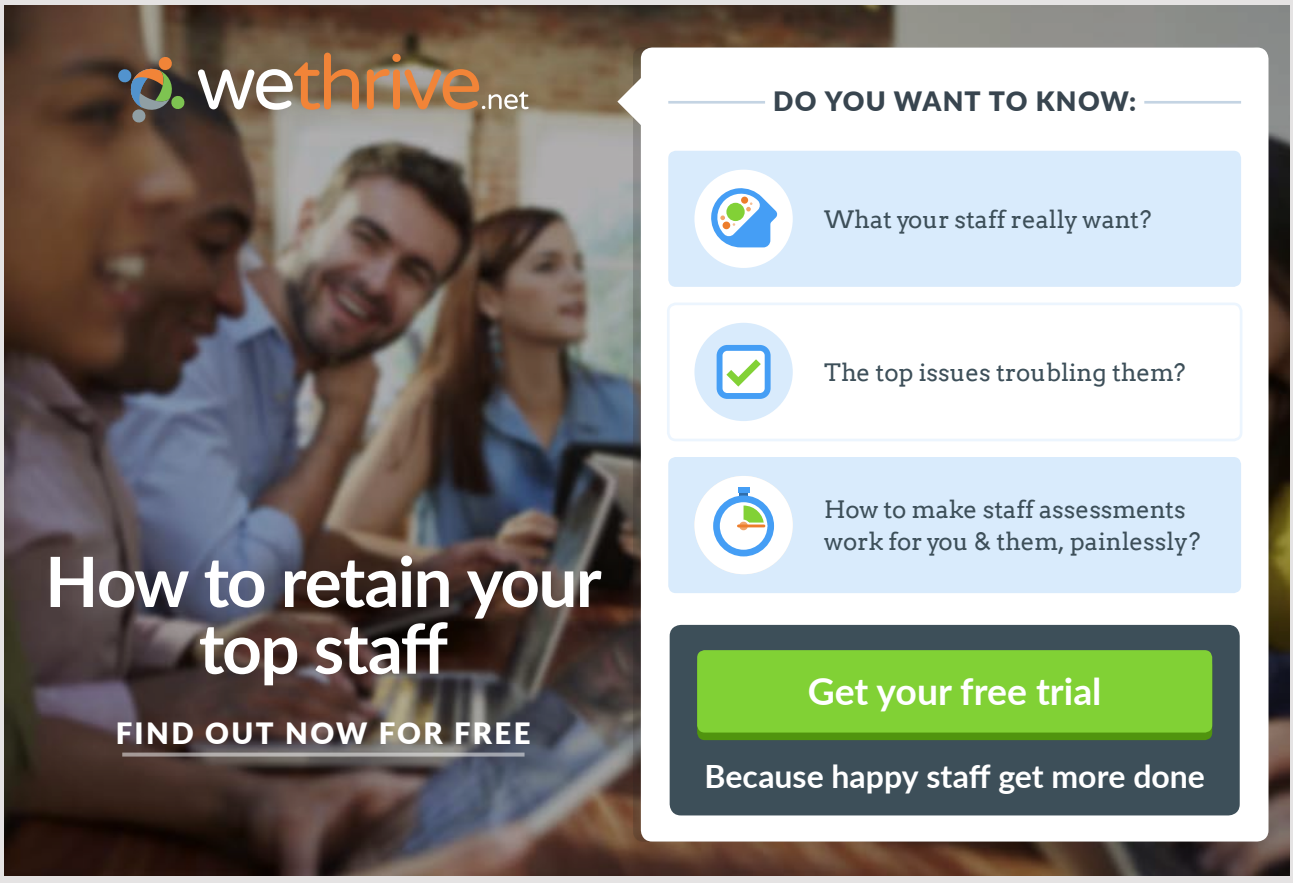
Let: $A_1 := 1$ $A_2 := 1$ m^2

$T_1 := 300$ K $T_2 := 22$ K

$p := 0.12$ Pa

$R_{gas} := 2077$ J/kg.K

$\gamma := 1.667$ ..ratio of sp. heats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Accommodation coefficients for helium ...from data:

$a_1 := 0.3$ accommodation coeff for helium at 300 K

$a_2 := 0.6$ accommodation coeff for helium at 22 K

$a_s := 0.4$ accommodation coeff for the shield

Noting that accommodation coeff factor, Fa has the same form as that of emissivity factor, for parallel plates, we write:

$$\frac{1}{F_a} = \left(\frac{1}{a_1} + \frac{1}{a_s} - 1\right) + (N-1)\cdot\left(\frac{2}{a_s} - 1\right) + \left(\frac{1}{a_2} + \frac{1}{a_s} - 1\right)$$ **...when there are N shields between the warm and cold plates**

When there is only one shield, we write:

$$\frac{1}{F_a} = \left(\frac{1}{a_1} + \frac{1}{a_s} - 1\right) + \left(\frac{1}{a_2} + \frac{1}{a_s} - 1\right)$$

i.e. $$\frac{1}{F_a} = \frac{1}{a_1} + \frac{1}{a_2} + 2\cdot\left(\frac{1}{a_s} - 1\right)$$

i.e. $$F_a := \frac{1}{\left[\frac{1}{a_1} + \frac{1}{a_2} + 2\cdot\left(\frac{1}{a_s} - 1\right)\right]}$$ i.e. $F_a = 0.125$

And, $$G := \frac{\gamma + 1}{\gamma - 1}\cdot\left(\frac{R_{gas}}{8\cdot\pi\cdot T_1}\right)^{\frac{1}{2}}\cdot F_a$$ i.e. $G = 0.262$

Therefore, heat transfer by molecular gas conduction is given by:

$$Q_{gas} := G\cdot p\cdot A_1\cdot\left(T_1 - T_2\right)$$

i.e. $Q_{gas} = 8.751$ **W .... heat transferred from warm surface to cold surface, with a shield in between....Ans.**

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If the shield were not there, what would be the heat transfer rate ?

When there is no radiation shield, accommodation coeff factor, F_a would be:

$$F_a := \frac{1}{\frac{1}{a_1} + \frac{1}{a_2} - 1} \qquad \text{i.e.} \quad F_a = 0.25$$

and,

$$G := \frac{\gamma + 1}{\gamma - 1} \cdot \left(\frac{R_{gas}}{8 \cdot \pi \cdot T_1} \right)^{\frac{1}{2}} \cdot F_a \qquad \text{i.e.} \quad G = 0.525$$

Therefore, heat transfer by molecular gas conduction when there is *no shield* is given by:

$$Q_{gas} := G \cdot p \cdot A_1 \cdot (T_1 - T_2)$$

i.e. $Q_{gas} = 17.503$ **W .... heat transferred from warm surface to cold surface, with no shield in between....Ans.**

==========

Prob.6.22. Determine the mean apparent thermal conductivity of a multilayer insulation (a) between 300 K (540°R) and 20.5 K (36.9°R), and (b) between 20.5 K (36.9°R) and 4.4 K (7.9°R), if the insulation is made up of 25 layers/cm (63.5 layers/in.) of aluminium foil (emissivity = 0.05) and fiberglass paper (h_c = 85 mW/m²-K = 0.015 Btu/hr-ft²-°F).[1]

Solution:

We have: for a well-evacuated multilayer insulation, apparent thermal conductivity may be calculated from:

$$k_t = \left(\frac{N}{\Delta x} \right)^{-1} \cdot \left[h_c + \frac{\sigma \cdot e \cdot \left(T_h^2 + T_c^2 \right) \cdot \left(T_h + T_c \right)}{2 - e} \right]$$

where:

h_c = solid conductance for spacer material

σ = Stefan - Boltzmann constant

e = effective emissivity of shield material

T_h, T_c = boundary temperatures of the insulation

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For this problem, we have:

$h_c := 85 \cdot 10^{-3}$ W/m^2.K

$Nby\Delta x := 25 \cdot 100$ layers/m

$\sigma := 5.669 \cdot 10^{-8}$ W/m^2.K^4 ..... Stefan-Boltzmann const

$e := 0.05$ effective emissivity of shield material

$T_h := 300$ K

$T_c := 20.5$ K

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Then, we get:

$$k_t := (N by \Delta x)^{-1} \cdot \left[h_c + \frac{\sigma \cdot e \cdot \left(T_h^{\,2} + T_c^{\,2} \right) \cdot \left(T_h + T_c \right)}{2 - e} \right]$$

i.e. $k_t = 5.085 \cdot 10^{-5}$ W/m.K = 50.85 μW/m.K .....thermal cond. of this multilayer insulation....Ans.

(b) When the T_h, T_c are: 20.5 K and 4.4 K respectively:

$T_h := 20.5$ K

$T_c := 4.4$ K

Then, we get:

$$k_t := (N by \Delta x)^{-1} \cdot \left[h_c + \frac{\sigma \cdot e \cdot \left(T_h^{\,2} + T_c^{\,2} \right) \cdot \left(T_h + T_c \right)}{2 - e} \right]$$

i.e. $k_t = 3.401 \cdot 10^{-5}$ W/m.K = 34.01 μW/m.K ....

....thermal cond. of this multilayer insulation
when T_h = 22.5 K, T_c = 4.4 K....Ans.

==========

Prob.6.23. A vapor-shielded liquid-helium vessel has conductances of U2 = 5 mw/K between the outer surface and shield and U1 = 6 mW/K between the shield and inner vessel. The mean specific heat of the helium gas flowing through the shield is 5.392 kJ/kg-K, and the heat of vaporization of the helium is 19.69 kJ/kg. The inner-vessel temperature is 4.4 K (7.9°R), and the outer-vessel temperature is 300 K (540°R). Determine: (a) the temperature of the shield, and (b) the heat-transfer rate to the inner vessel.[1]

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Solution:

Data:

$T_1 := 4.4$ K .... temp of inner vessel

$T_2 := 300$ K .... temp of outer vessel

$U_1 := 6 \cdot 10^{-3}$ W/K ....conductance between shield and inner vessel

$U_2 := 5 \cdot 10^{-3}$ W/K ....conductance between outer vessel and the shield

$h_{fg} := 19.69 \cdot 10^{3}$ J/kg.... heat of vaporization of helium

$c_p := 5.392 \cdot 10^{3}$ J/kg.K....mean sp. heat of helium gas flowing through the shield

For vapor shielded vessels, Dimensionless temp of the shield is given by:

$$\theta = \frac{\pi 2 + 1}{2 \cdot \pi 1 \cdot \pi 2} \cdot \left[\left[1 + \frac{4 \cdot \pi 1 \cdot \pi 2}{(\pi 2 + 1)^2} \right]^{\frac{1}{2}} - 1 \right]$$

where:

$$\pi 1 := \frac{c_p \cdot (T_2 - T_1)}{h_{fg}} \qquad \text{i.e} \quad \pi 1 = 80.948$$

$$\pi 2 := \frac{U_1}{U_2} \qquad \text{i.e.} \quad \pi 2 = 1.2$$

Therefore:

$$\theta := \frac{\pi 2 + 1}{2 \cdot \pi 1 \cdot \pi 2} \cdot \left[\left[1 + \frac{4 \cdot \pi 1 \cdot \pi 2}{(\pi 2 + 1)^2} \right]^{\frac{1}{2}} - 1 \right] \qquad \text{where} \quad \theta = \frac{T_s - T_1}{T_2 - T_1}$$

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We get:

i.e. $\theta = 0.091$

Therefore, temp of the shield, T_s is:

$$T_s := \theta \cdot (T_2 - T_1) + T_1$$

i.e. $T_s = 31.231$ **K ..... temp of shield ... Ans.**

Heat transfer rate to the inner vessel, Q_s-1:

$$Q\_s1 := U_1 \cdot (T_s - T_1) \quad W$$

i.e. $Q\_s1 = 0.161$ **W.... heat transfer rate to inner vessel... Ans.**

==========

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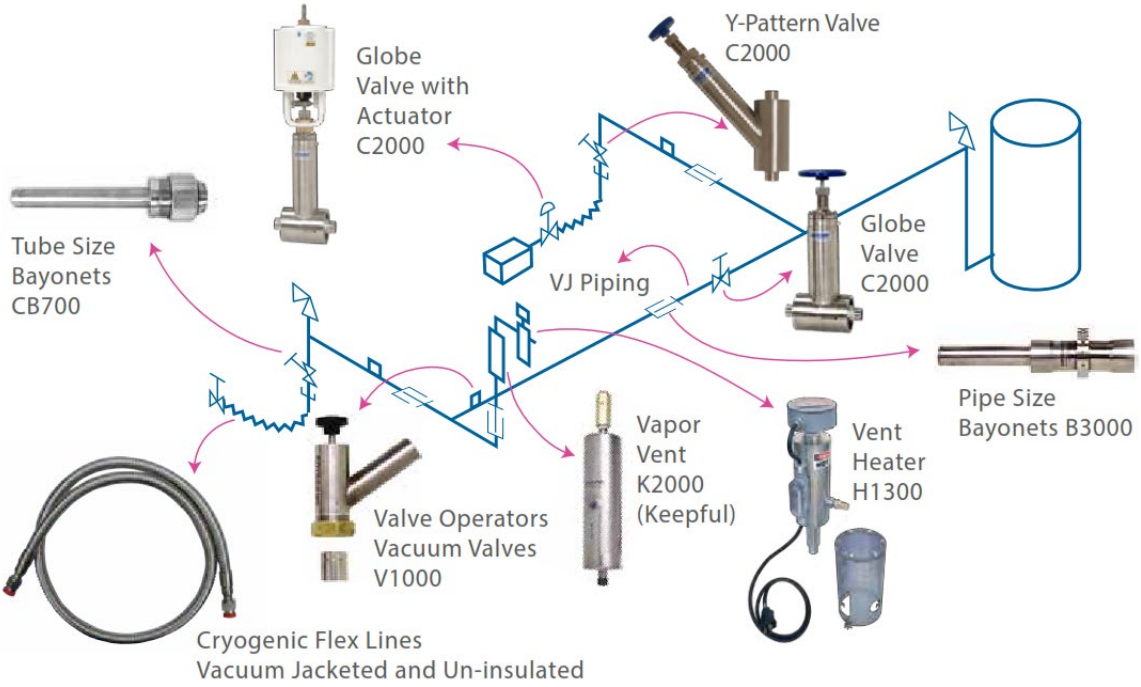
6.5 VACUUM-INSULATED TRANSFER LINES [1]

Fig. 6.12. Various components in a vacuum-insulated piping lay out are shown below:

(Ref: www.Cryofab.com):



Vacuum insulated line consists of an inner line in which the cryogenic fluid flows, and an outer vacuum jacket. It may be only vacuum or a multilayer insulation with vacuum.

Typical heat leaks for Cryofab vacuum insulated lines are shown below, as an example:

CRYOFAB VACUUM INSULATED PIPE SPECIFICATIONS					
Vacuum Insulated Pipe Size				Heat Leak (BTU per Hour)	
Inner Pipe	Outer Pipe	Outer Diameter	Wt./Ft.	Pipe	Bayonet
.5"	2.0"	2.375"	2.4 lb	.32/ft	9.5
1.0"	2.5"	2.875"	3.7 lb	.42/ft	14.0
1.5"	3.0"	3.5"	4.7 lb	.54/ft	12.9
2.0"	3.5"	4.0"	5.6 lb	.74/ft	18.6

All pipe sizes shown above are Type 304 Stainless Steel Sch 5 Pipe.
MAWP (Maximum Allowable Working Pressure) of the inner pipe is 150PSI for products listed.

Design of vacuum insulated lines: The vacuum insulated lines are designed as per ASA code for pressure piping.

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Inner line is designed to withstand the necessary internal pressure, using the same equation as used for a piping (see section 6.2.5), viz;

$$t = p \cdot \frac{D_o}{2 \cdot s_a + 0.8 \cdot p}$$

where: p = design pressure, Do = outside dia of pipe, sa = allowable stress of pipe material.

Outer line must be designed for collapsing under external pressure of atmosphere. For this, equations given under section 6.2.5 for a cylindrical vacuum vessel may be used.

Recollect:

For a **long cylinder**, following Length to Diameter ratio condition must be met:

$$\mathbf{L/D_o > 1.140\ (1 - \upsilon^2)^{1/4}\ (Do/t)^{1/2}}$$

where: L is the unsupported length (distance between stiffening rings for outer shell) of the cylinder.

The collapsing or critical pressure for a *long cylinder* exposed to external pressure is:

$$p_c = \frac{2 \cdot E \cdot \left(\frac{t}{D_0}\right)^3}{1 - \nu^2}$$

where:

E = Young's modulus of shell material; t = shell thickness

D_0 = outside dia of shell; υ = Poisson's ratio for shell material

The collapsing or critical pressure for a *short cylinder* exposed to external pressure is:

$$p_c = \frac{2.42 \cdot E \cdot \left(\frac{t}{D_0}\right)^{\frac{5}{2}}}{\left(1 - \nu^2\right)^{\frac{3}{4}} \cdot \left[\left(\frac{L}{D_0}\right) - 0.45 \cdot \left(\frac{t}{D_0}\right)^{\frac{1}{2}}\right]}$$

where: L is the distance between stiffening rings for outer shell.

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Thermal contraction: This is a problem in long cryogenic lines. Thermal contraction is taken care by providing expansion bellows or U-bends in the lines. Bellows in the inner lines should be avoided, since maintenance will be difficult if the bellows develop any leaks. So, for inner lines, flexibility may be achieved by using U-bends. See Fig. below:

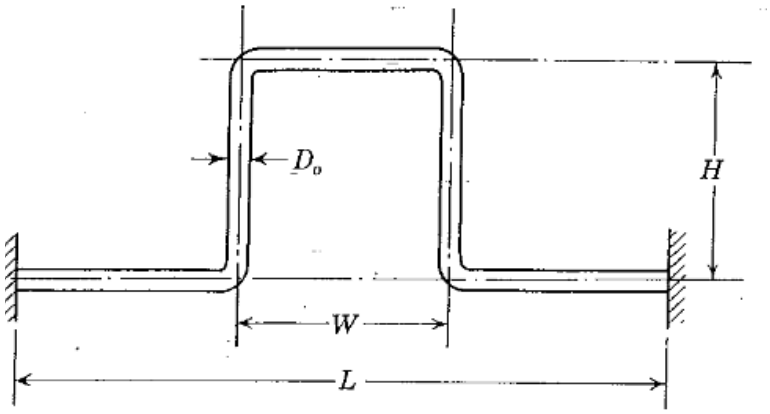

Fig. 6.13. U-bend

Maximum stress induced by thermal contraction in a U-bend is given by:

For α = W/L >=1/2:

$$\frac{\sigma_{max} \cdot L}{E \cdot e_t \cdot D_o} = \frac{\alpha + \beta}{B} \cdot \left[1 + \frac{(1 + 2 \cdot \beta) \cdot D_o}{4 \cdot (\alpha + \beta) \cdot H} \right]$$

For α = W/L < 1/2:

$$\frac{\sigma_{max} \cdot L}{E \cdot e_t \cdot D_o} = \frac{1 - \alpha + \beta}{B} \cdot \left[1 + \frac{(1 + 2 \cdot \beta) \cdot D_o}{4 \cdot (1 - \alpha + \beta) \cdot H} \right]$$

where:

$$B = \frac{2}{3} \cdot \beta \cdot \left[\beta \cdot (2 + \beta) + 3 \cdot \alpha \cdot (1 - \alpha) + \frac{3}{8} \cdot (1 + 2 \cdot \beta) \cdot \left(\frac{D_0}{H} \right)^2 \right]$$

$$\alpha = \frac{W}{L} \qquad \beta = \frac{H}{L}$$

$$e_t = \int_{T_c}^{T_h} \lambda_t dT \quad = \text{unit thermal strain, see Table 6.16}$$

D_o = outside dia of line

E = Young's Modulus

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Optimum value of ratio α = W/L = 0.5; i.e. for a fixed value of β = H/L, α = 0.5 gives minimum thermal stress.

Following Table 6.16 gives unit thermal strain $e_t = (\Delta L/L)$ for several solids.

Table 6.16 Unit thermal expansion for several solids (Corruccini and Gniewek 1961).
$e_t = \Delta L/L = \int_0^T \lambda_t \, dT$; multiply the numbers in the table by 10^{-5}

Temperature (K)	Beryllium Copper	Aluminum	1020 Steel	304 Stainless	Monel	Invar	Yellow Brass	Plexiglass	Teflon	Pyrex Glass	Nylon	Polystyrene
0	0	0	0	0	0	0	0	0	0	0	0	0
20	0	0	0	0	0	0	1	15	30	−1.0	10	28
40	1	2	1	0	1	0	4	60	80	−2.0	37	84
50	3	5	2	2	3	0	9	83	109	−1.9	58	118
60	7	10	4	5	6	0	16	110	140	−1.5	81	156
70	12	16	7	11	10	1	24	136	176	−0.6	110	196
80	20	24	10	17	15	2	34	170	210	+1.0	142	242
90	29	34	15	25	21	3	45	196	250	2.8	177	286
100	39	45	20	35	28	5	58	230	290	4.5	217	339
120	61	72	32	55	45	9	85	290	380	8.5	301	445
140	85	103	47	78	64	13	115	360	480	13.0	393	558
160	110	138	64	103	84	18	147	440	600	17.5	493	676
180	137	175	82	129	107	23	180	530	740	22.5	600	798
200	165	214	101	157	130	29	215	630	900	27.5	716	924
250	242	318	155	229	193	41	304	915	1390	41.7	1050	1250
300	329	431	210	307	261	54	397	1275	1600	57.0	1450	1601

Table 6.16 Unit Thermal strain for various solids [1]
(multiply the numbers in Table by 10^{-5})

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Generally, we are interested in Thermal expansion between the warm end at a temp. of TH and the cold end at a temp of TL, for a given material, to calculate the thermal stress.

Let us write the required Mathcad Functions for a few materials:

First, copy the data for a few materials in Table 6.16 in to a Matrix, MX:

MX :=

T(K)	BeCu	AL 1020	Steel SS 304	Monel	Invar	Teflon	Pyrex	Nylon	Polysty--rene
0	0	0	0	0	0	0	0	0	0
20	0	0	0	0	0	30	-1	10	28
40	1	2	1	0	1	0	80	-2	37	84
50	3	5	2	2	3	0	109	-1.9	58	118
60	7	10	4	5	6	0	140	-1.5	81	156
70	12	16	7	11	10	1	176	-0.6	110	196
80	20	24	10	17	15	2	210	1	142	242
90	29	34	15	25	21	3	250	2.8	177	286
100	39	45	20	35	26	5	290	4.5	217	339
120	61	72	32	55	45	9	380	8.5	301	445
140	85	103	47	78	64	13	480	13	393	558
160	110	138	64	103	84	18	600	17.5	493	676
180	137	175	82	129	107	23	740	22.5	600	798
200	165	214	101	157	130	29	900	27.5	716	924
250	242	318	155	229	193	41	1390	41.7	1050	1250
300	329	431	210	307	261	54	1600	57	1450	1601

$Temp := MX^{<0>}$

Temp =

	0
0	0
1	20
2	40
3	50
4	60
5	70
6	80
7	90
8	100
9	120
10	140
11	160
12	180
13	200
14	250

First column gives temp in K, and other columns give values of (ΔL/L)*10^5 for some materials as shown.

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Now, write the desired Mathcad Functions to find (ΔL/L) for different materials, between TH and TL:

For BeCu:

$$\Delta LbyL\_BeCu\_TH\_to\_TL(TH, TL) := \begin{cases} \text{return "TL, TH must be between 0 K and 300 K!"} & \text{if } TL < 0 \\ \text{return "TL, TH must be between 0 K and 300 K!"} & \text{if } TH > 300 \\ \text{linterp}\left[Temp, (MX)^{\langle 1 \rangle}, TH\right] \cdot 10^{-5} - \text{linterp}\left[Temp, (MX)^{\langle 1 \rangle}, TL\right] \cdot 10^{-5} \end{cases}$$

Ex: $\Delta LbyL\_BeCu\_TH\_to\_TL(300, 90) = 3 \cdot 10^{-3}$ m/m

---

For ALuminium:

$$\Delta LbyL\_AL\_TH\_to\_TL(TH, TL) := \begin{cases} \text{return "TL, TH must be between 0 K and 300 K!"} & \text{if } TL < 0 \\ \text{return "TL, TH must be between 0 K and 300 K!"} & \text{if } TH > 300 \\ \text{linterp}\left[Temp, (MX)^{\langle 2 \rangle}, TH\right] \cdot 10^{-5} - \text{linterp}\left[Temp, (MX)^{\langle 2 \rangle}, TL\right] \cdot 10^{-5} \end{cases}$$

Ex: $\Delta LbyL\_AL\_TH\_to\_TL(300, 90) = 3.97 \cdot 10^{-3}$ m/m

---

For Steel1020:

$$\Delta LbyL\_Steel1020\_TH\_to\_TL(TH, TL) := \begin{cases} \text{return "TL, TH must be between 0 K and 300 K!"} & \text{if } TL < 0 \\ \text{return "TL, TH must be between 0 K and 300 K!"} & \text{if } TH > 300 \\ \text{linterp}\left[Temp, (MX)^{\langle 3 \rangle}, TH\right] \cdot 10^{-5} - \text{linterp}\left[Temp, (MX)^{\langle 3 \rangle}, TL\right] \cdot 10^{-5} \end{cases}$$

Ex: $\Delta LbyL\_Steel1020\_TH\_to\_TL(300, 90) = 1.95 \cdot 10^{-3}$ m/m

---

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For SS304:

$$\Delta LbyL\_SS304\_TH\_to\_TL(TH, TL) := \begin{cases} \text{return "TL, TH must be between 0 K and 300 K!"} & \text{if } TL < 0 \\ \text{return "TL, TH must be between 0 K and 300 K!"} & \text{if } TH > 300 \\ \text{linterp}\left[Temp, (MX)^{\langle 4 \rangle}, TH\right] \cdot 10^{-5} - \text{linterp}\left[Temp, (MX)^{\langle 4 \rangle}, TL\right] \cdot 10^{-5} \end{cases}$$

Ex: $\Delta LbyL\_SS304\_TH\_to\_TL(300, 90) = 2.82 \cdot 10^{-3}$ m/m

---

For Monel:

$$\Delta LbyL\_Monel\_TH\_to\_TL(TH, TL) := \begin{cases} \text{return "TL, TH must be between 0 K and 300 K!"} & \text{if } TL < 0 \\ \text{return "TL, TH must be between 0 K and 300 K!"} & \text{if } TH > 300 \\ \text{linterp}\left[Temp, (MX)^{\langle 5 \rangle}, TH\right] \cdot 10^{-5} - \text{linterp}\left[Temp, (MX)^{\langle 5 \rangle}, TL\right] \cdot 10^{-5} \end{cases}$$

Ex: $\Delta LbyL\_Monel\_TH\_to\_TL(300, 90) = 2.4 \cdot 10^{-3}$ m/m

---

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For Invar:

$$\Delta LbyL\_Invar\_TH\_to\_TL(TH, TL) := \begin{cases} \text{return "TL, TH must be between 0 K and 300 K!"} & \text{if } TL < 0 \\ \text{return "TL, TH must be between 0 K and 300 K!"} & \text{if } TH > 300 \\ \text{linterp}\left[Temp, (MX)^{\langle 6 \rangle}, TH\right] \cdot 10^{-5} - \text{linterp}\left[Temp, (MX)^{\langle 6 \rangle}, TL\right] \cdot 10^{-5} \end{cases}$$

Ex: $\Delta LbyL\_Invar\_TH\_to\_TL(300, 90) = 5.1 \cdot 10^{-4}$ m/m

---

For Teflon:

$$\Delta LbyL\_Teflon\_TH\_to\_TL(TH, TL) := \begin{cases} \text{return "TL, TH must be between 0 K and 300 K!"} & \text{if } TL < 0 \\ \text{return "TL, TH must be between 0 K and 300 K!"} & \text{if } TH > 300 \\ \text{linterp}\left[Temp, (MX)^{\langle 7 \rangle}, TH\right] \cdot 10^{-5} - \text{linterp}\left[Temp, (MX)^{\langle 7 \rangle}, TL\right] \cdot 10^{-5} \end{cases}$$

Ex: $\Delta LbyL\_Teflon\_TH\_to\_TL(300, 90) = 0.0135$ m/m

For Pyrex:

$$\Delta LbyL\_Pyrex\_TH\_to\_TL(TH, TL) := \begin{cases} \text{return "TL, TH must be between 0 K and 300 K!"} & \text{if } TL < 0 \\ \text{return "TL, TH must be between 0 K and 300 K!"} & \text{if } TH > 300 \\ \text{linterp}\left[Temp, (MX)^{\langle 8 \rangle}, TH\right] \cdot 10^{-5} - \text{linterp}\left[Temp, (MX)^{\langle 8 \rangle}, TL\right] \cdot 10^{-5} \end{cases}$$

Ex: $\Delta LbyL\_Pyrex\_TH\_to\_TL(300, 90) = 5.42 \cdot 10^{-4}$ m/m

---

For Nylon:

$$\Delta LbyL\_Nylon\_TH\_to\_TL(TH, TL) := \begin{cases} \text{return "TL, TH must be between 0 K and 300 K!"} & \text{if } TL < 0 \\ \text{return "TL, TH must be between 0 K and 300 K!"} & \text{if } TH > 300 \\ \text{linterp}\left[Temp, (MX)^{\langle 9 \rangle}, TH\right] \cdot 10^{-5} - \text{linterp}\left[Temp, (MX)^{\langle 9 \rangle}, TL\right] \cdot 10^{-5} \end{cases}$$

Ex: $\Delta LbyL\_Nylon\_TH\_to\_TL(300, 90) = 0.01273$ m/m

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For Polystyrene:

$$\Delta LbyL\_Polystyrene\_TH\_to\_TL(TH, TL) := \begin{cases} \text{return "TL, TH must be between 0 K and 300 K!"} & \text{if } TL < 0 \\ \text{return "TL, TH must be between 0 K and 300 K!"} & \text{if } TH > 300 \\ \text{linterp}\left[\text{Temp}, (MX)^{\langle 10 \rangle}, TH\right] \cdot 10^{-5} - \text{linterp}\left[\text{Temp}, (MX)^{\langle 10 \rangle}, TL\right] \cdot 10^{-5} \end{cases}$$

Ex: $\Delta LbyL\_Polystyrene\_TH\_to\_TL(300, 90) = 0.01315\,m/m$

Prob.6.24. A vacuum-jacketed liquid hydrogen transfer line has a 300-mm nominal (12-in. nominal) inner line and a 400-mm nominal (16-in. nominal) outer 1ine. Both lines are constructed of 304 stainless steel, and the weld efficiency is 100 percent. The outer line has a distance of 12.25 m (40.2 ft) between support points. Determine the minimum thickness and schedule number for the inner and outer lines, for a design internal pressure of 600 kPa abs.[1]

Solution

From Table 6.9, for the 300 mm nominal inner pipe, the OD is 323.9 mm, and for the 400 mm nominal outer pipe, the OD is 406.4 mm.

From Table 6.1, allowable stress for SS 304 is 129.2 MPa.

Then, minimum thickness of inner pipe is determined from:

For piping subjected to internal pressure:

$$t = p \cdot \frac{D_o}{2 \cdot s_a + 0.8 \cdot p} \qquad \text{....thickness for piping}$$

where: p = design pressure, Do = outside dia of pipe, s_a = allowable stress of pipe material.

For the inner pipe, we have:

$D_o := 323.9 \quad mm$

$p := 600 \quad kPa$

$s_a := 129.2 \cdot 10^3 \quad kPa$

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Then, thickness of inner pipe is:

$$t := p \cdot \frac{D_o}{2 \cdot s_a + 0.8 \cdot p} \qquad \text{i.e.} \quad t = 0.751 \quad \text{mm}$$

Now, from Table 6.9, we see that: 300 mm nominal, schedule 5 pipe has a thickness of 4 mm.

Therefore, 300 mm, schedule 5 pipe can be used for inner pipe.

Min. thickness of outer pipe:

First, assume that it is a 'long' pipe, and calculate the thickness. Then, verify that it is really a long pipe, by checking the criterion for a long pipe.

For a long pipe, subjected to buckling from outside atm. pressure:

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$p_a := 101.3$ kPa .... atm. pressure

$v := 0.28$ ...Poisson's ratio for SS 304

$E := 207 \cdot 10^6$ kPa... young's Modulus

$D_o := 406.4$ mm ... OD of outer pipe

Then, thickness is:

$$t := \left[\frac{p_c \cdot \left(1 - v^2\right)}{2 \cdot E}\right]^{\frac{1}{3}} \cdot D_o \qquad \text{i.e} \quad t = 3.927 \quad \text{mm}$$

Again, from Table 6.9, we see that for 400 nominal pipe, sch 5 has a thickness of 4.2 mm.

Therefore, choose 400 nominal, schedule 5 pipe for outer pipe.

Now, verify the criterion for a 'long' pipe is satisfied:

The criterion for a long pipe is:

$$\frac{L}{D_0} > 1.14 \cdot \left(1 - v^2\right)^{\frac{1}{4}} \cdot \left(\frac{D_o}{t}\right)^{\frac{1}{2}}$$

We have:

$L := 12.25$ m .... length between supports for the outer pipe

$D_o := 406.4 \cdot 10^{-3}$ m .... ODof pipe

$t := 4.2 \cdot 10^{-3}$ m... thickness of pipe

Now:

$$1.14 \cdot \left(1 - v^2\right)^{\frac{1}{4}} \cdot \left(\frac{D_o}{t}\right)^{\frac{1}{2}} = 10.987$$

$\frac{L}{D_o} = 30.143$ **> 10.987 ... therefore the pipe is a 'long' cylinder.**

==========

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Prob.6.25. Determine the maximum thermal stress induced in a U-bend section, if the line is initially at 300 K (540°R) in the stress-free condition, and is cooled to 80 K (l44°R). The line is a 150-mm nominal (6-in. nominal) schedule 5 stainless steel pipe, with a length of 12 m (39.4 ft) between fixed points. The U-bend has dimensions of W = 6 m (19.7 ft) and H = 3 m (9.8 ft). Compare this value of stress with the thermal stress that would be induced if a 12 m length of straight pipe of the same diameter with both ends rigidly fixed were cooled through the same temperature range.[1]

Solution:

We have:

$W := 6$ m .... width of U-bend

$H := 3$ m .... height of U-bend

$L := 12$ m .... distance between fixed points

$D_o := 168.3 \cdot 10^{-3}$ m.....OD of 150 mm nominal pipe

$\nu := 0.28$ ...Poisson's ratio for SS 304

$E := 207 \cdot 10^{6}$ kPa... young's Modulus

$e_t := (307 - 17) \cdot 10^{-5}$ = unit thermal strain, ΔL/L for SS 304 between 300 K and 80 K... from Table 6.16

OR: Using the Mathcad Function written above for (ΔL/L) = e_t values for SS 304, we get:

For SS304:

$$\Delta LbyL\_SS304\_TH\_to\_TL(TH, TL) := \begin{cases} \text{return "TL, TH must be between 0 K and 300 K!" if } TL < 0 \\ \text{return "TL, TH must be between 0 K and 300 K!" if } TH > 300 \\ \text{linterp}\left[Temp, (MX)^{\langle 4 \rangle}, TH\right] \cdot 10^{-5} - \text{linterp}\left[Temp, (MX)^{\langle 4 \rangle}, TL\right] \cdot 10^{-5} \end{cases}$$

i.e. $\Delta LbyL\_SS304\_TH\_to\_TL(300, 80) = 2.9 \cdot 10^{-3}$ m/m

First, calculate dimensionless parameters α and β:

$\alpha := \frac{W}{L}$ i.e. $\alpha = 0.5$

$\beta := \frac{H}{L}$ i.e. $\beta = 0.25$

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For α = W/L >=1/2, we determine the max. thermal stress from:

$$\frac{\sigma_{max} \cdot L}{E \cdot e_t \cdot D_o} = \frac{\alpha + \beta}{B} \cdot \left[1 + \frac{(1 + 2 \cdot \beta) \cdot D_o}{4 \cdot (\alpha + \beta) \cdot H} \right] \qquad \text{where:}$$

$$B := \frac{2}{3} \cdot \beta \cdot \left[\beta \cdot (2 + \beta) + 3 \cdot \alpha \cdot (1 - \alpha) + \frac{3}{8} \cdot (1 + 2 \cdot \beta) \cdot \left(\frac{D_0}{H} \right)^2 \right]$$

Therefore, B = 0.358

And,

$$\frac{\alpha + \beta}{B} \cdot \left[1 + \frac{(1 + 2 \cdot \beta) \cdot D_o}{4 \cdot (\alpha + \beta) \cdot H} \right] = 2.152$$

$$\sigma_{max} := \left[\frac{\alpha + \beta}{B} \cdot \left[1 + \frac{(1 + 2 \cdot \beta) \cdot D_o}{4 \cdot (\alpha + \beta) \cdot H} \right] \right] \cdot \frac{E \cdot e_t \cdot D_o}{L}$$

i.e. $\sigma_{max} = 1.812 \cdot 10^4$ **kPa = 18.12 MPa ..Max. thermal stress induced... Ans.**

(b) If a 12 m length of straight pipe were used, without an U-bend, with the ends rigidly fixed, what would be the thermal stress?

Between 300 K and 80 K, the thermal contraction ΔL is (from Table 6.16):

$$\Delta L := e_t \cdot L$$

i.e. ΔL = 0.035 **m = 35 mm ... Ans.**

And, the thermal stress = Young's Modulus x strain

i.e. $\sigma_{max\_with\_No\_Ubend} := E \cdot e_t$

i.e. $\sigma_{max\_with\_No\_Ubend} = 6.003 \cdot 10^5$ **kPa**

i.e. ....max. thermal stress when there is no U-bend = 600.3 MPa.... Ans.

==========

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6.6 VACUUM-INSULATED LINE JOINTS

When lengths more than 12 m are used, it is necessary to use joints in the vacuum jacketed lines. These are known as 'Bayonet joints', developed by Johnston in 1949, and are convenient when low heat leaks are required and frequent dismantling of joints are required. It consists of a male portion telescoping in to a female portion, with a very small clearance in between the two portions so that no liquid flows in the gap, thus suppressing the gas-convection. Sealing problem is reduced by having a neoprene O ring seal in the warm flange rather than at the cold end. For LH2 transfer line joints, typical heat in-leaks vary from about 3.2 W for 80 mm x 125 mm joints to about 8.8 W for 250 mm x 300 mm joints.

Typical commercial Bayonet joint from Cryocomp, a division of Cryofab (Ref: www.cryocomp.com) is shown below, as an example:

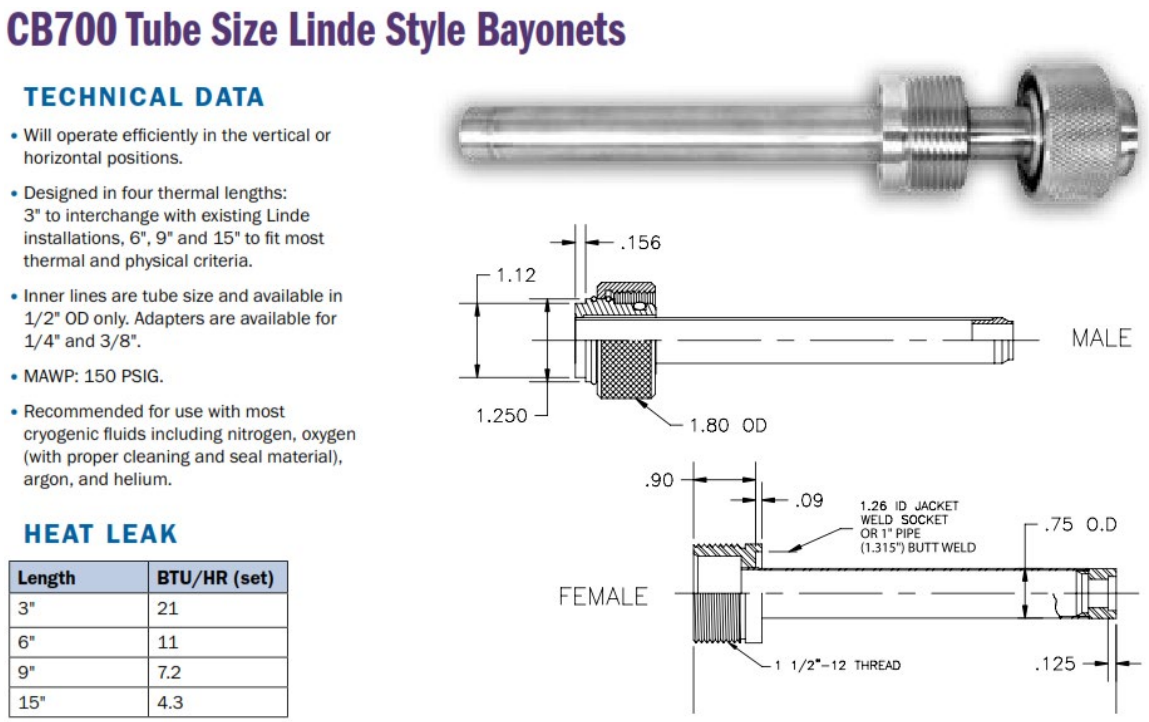
CB700 Tube Size Linde Style Bayonets
TECHNICAL DATA
• Will operate efficiently in the vertical or horizontal positions.
• Designed in four thermal lengths: 3" to interchange with existing Linde installations, 6", 9" and 15" to fit most thermal and physical criteria.
• Inner lines are tube size and available in 1/2" OD only. Adapters are available for 1/4" and 3/8".
• MAWP: 150 PSIG.
• Recommended for use with most cryogenic fluids including nitrogen, oxygen (with proper cleaning and seal material), argon, and helium.
HEAT LEAK
Length | BTU/HR (set)
3" | 21
6" | 11
9" | 7.2
15" | 4.3
.156
1.12
1.250
1.80 OD
MALE
.90
.09
1.26 ID JACKET
WELD SOCKET
OR 1" PIPE
(1.315") BUTT WELD
.75 O.D
FEMALE
1 1/2"-12 THREAD
.125

Fig. 6.14. Various Bayonet joint for vacuum insulated lines

Note: 1 BTU/hr = 0.2931 W; or, 1 W = 3.4121 BTU/hr)

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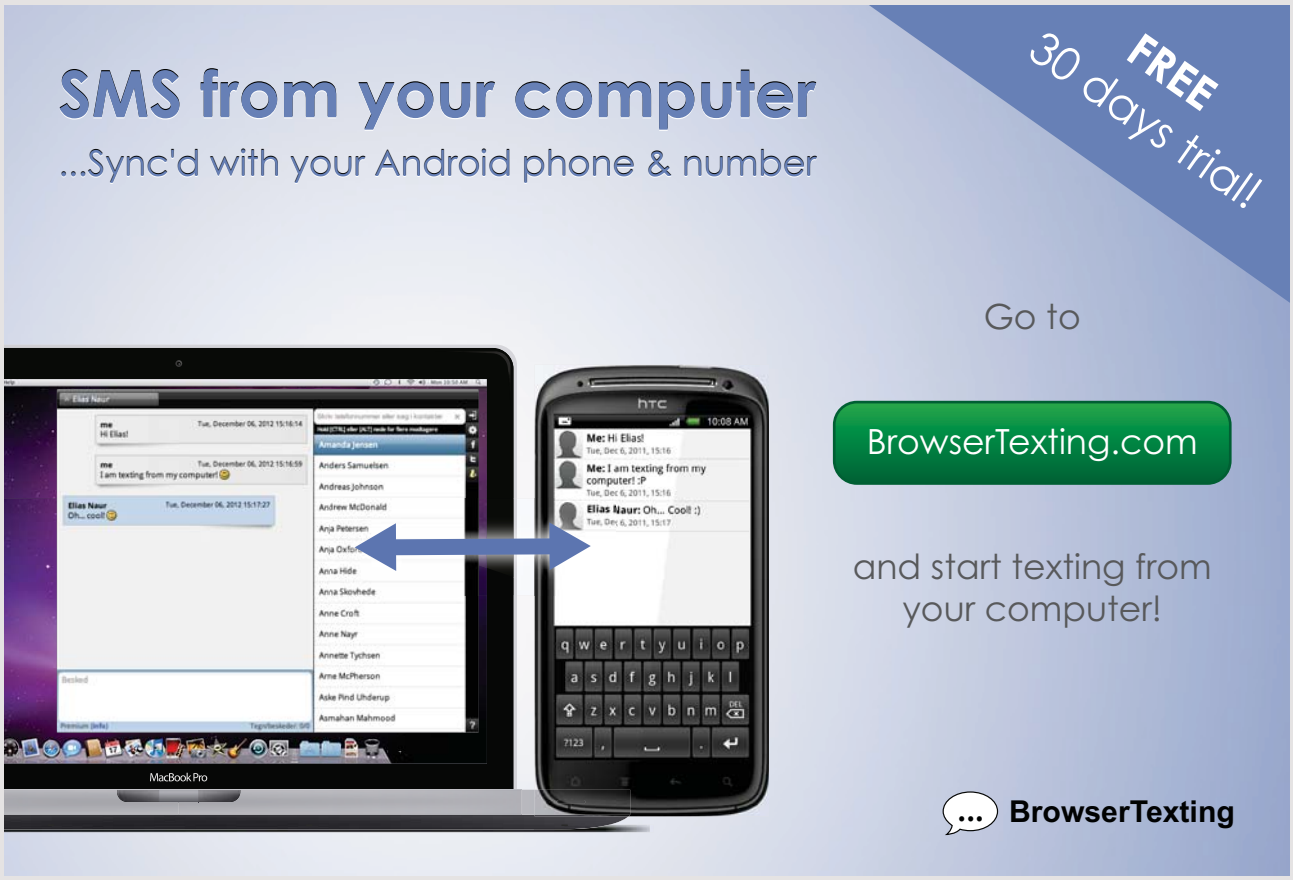
6.7 CRYOGENIC VALVES

These are of two types: (i) Extended stem valves, and (ii) Vacuum jacketed valves.
Extended stem type is like an ordinary valve, but has a long stem, 250 mm to 300 mm of thin walled tubing. So, the valve handle is at nearly ambient temperature, and handling and sealing problem are minimized.
Vacuum jacketed type of valve also has a long stem, but with a vacuum jacket around it, so that heat leak is very much reduced. Many times, multilayer insulation may also be used to reduce heat in-leaks.

Some examples of cryogenic valves are shown below:

Typical long stem globe valve for -196^0 C applications from KITZ corpn.
Ref: https://www.kitz.co.jp/english2/pdf/E-426-05.pdf



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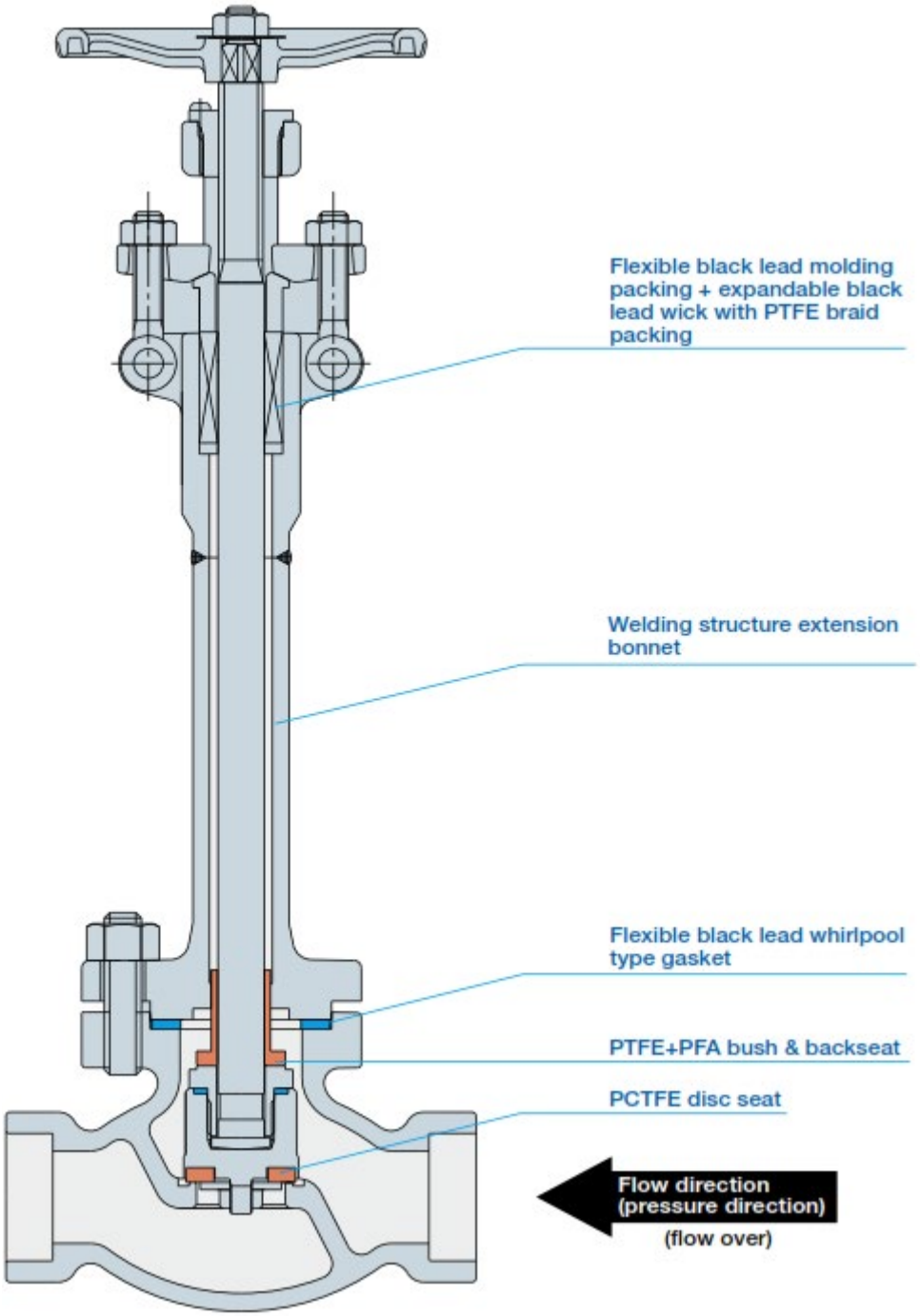
Flexible black lead molding packing + expandable black lead wick with PTFE braid packing
Welding structure extension bonnet
Flexible black lead whirlpool type gasket
PTFE+PFA bush & backseat
PCTFE disc seat
Flow direction (pressure direction)
(flow over)

Fig. 6.15. Typical long-stem cryogenic valve

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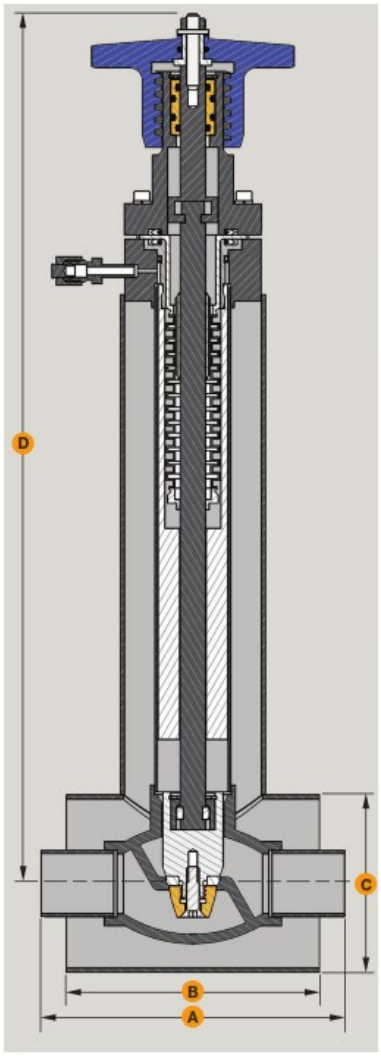
And, an example of Vacuum jacketed valve from Acme Cryogenics (www.acmecryo.com):



MODEL CV DIMENSION TABLE					
Size	Pipe Ends A	Tube Ends A	B	C	(Open) D
½"	4.75"	8.5"	3.75"	2.875"	17.8"
1"	6"	9"	5"	3.5"	17.8"
1½"	8"	10"	6"	4.5"	20.2"
2"	8.5"	11"	6.5"	5.563"	21.9"
3"	15.5"	TBD	13.25"	5.563"	25.25"
4"	15.5"	TBD	13.25"	6.625"	25.25"

PERFORMANCE CHARACTERISTICS VACUUM JACKETED VALVE				
Valve Size	Model No.	Flow Coefficient (Cv)	Conductive Heat Leak (BTU/HR)	Cool Down Weight (LBS)
½"	0500	6.6	6.4	2.6
1"	1000	14	9.2	3.3
1½"	1500	34	10.0	6.1
2"	2000	50	14.3	9.0
3"	3000	139	63.6	32.5
4"	4000	250	78.8	46.5

Fig. 6.16. Vacuum jacketed cryogenic valve

6.8 AMOUNT OF CRYOGENIC LIQUID REQUIRED TO COOL DOWN VACUUM INSULATED LINES, VALVES ETC

Basically, cooling with Liquid helium and/or liquid nitrogen is of interest. i.e. cooling from 300 K to 77 K or 4.2 K or cooling from 77 K to 4.2 K.

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Heat load can be calculated accurately using the Thermal conductivity integral values given earlier. However, for quick calculations in the laboratory, following method is suggested:

Information in following three Tables is from Ref.[12]:

Property	Nitrogen	Helium
Normal boiling point (NBP) (K)	77.3	4.22
Latent heat of vaporisation at NBP (Joules/gram)	198	20.9
Amount of liquid evaporated by 1 Watt at NBP (litre/hour)	0.024	1.38
Liquid density at NBP (g/cm^3)	0.808	0.125
Gas density at NTP (g/cm^3)	1.16×10^{-3}	1.66×10^{-4}
Gas at NTP to liquid at NBP volume ratio	694:1	750:1
Enthalpy change of gas (J/g). 4.2 K - 77 K	-	384
(at 1 atmosphere) 77 K - 273 K	234	1018

Table 6.17. Physical properties of Helium and Nitrogen

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Material	300 K to 0 K	77 K to 0 K
Copper (electrolytic)	1620	690
Aluminium (99% pure)	730	220
Copper (phosphor deoxidised)	460	54
Brass (typical)	170	20
Constantan	52	8.8
Stainless steel	30	3.2
G10 fibreglass (typical)	1.5	0.17
Nylon	0.8	0.13

Table 6.18 Thermal conductivity integrals of some common materials used in cryostats (W/cm)

Note: When the warm end is at 300 K or 77 K, and the cold end is at *20 K or below*, use the values from the above Table. But, if the conduction is from 300 K to 77 K, subtract the values given under (77K to 0K) from the value under (300K to 0K).

Then, heat conducted along a rod of length L, cross-sectional area A between temperature T_2 and T_1 is given by:

$$Q=\frac{A}{L}\int_{T_1}^{T_2} k(T)\,dT$$

The volume of liquid helium required to cool a mass of metal from one temperature to another using the full enthalpy of the gas or using the latent heat alone was reported by J B Jacobs, Advances in Cryogenic Engineering, Volume 8, 1963, p. 529, as follows.

Cryogen		^{4}He	^{4}He	N_2
Initial temperature of metal		300 K	77 K	300 K
Final temperature of metal		4.2 K	4.2 K	77 K
Using the latent heat	Al	66.6	3.20	1.01
of vaporisation only *	St. Steel	33.3	1.43	0.53
	Cu	31.1	2.16	0.46
Using the enthalpy	Al	1.61	0.22	0.64
of the gas **	St. Steel	0.79	0.11	0.33
	Cu	0.79	0.15	0.29

Table 6.19. Amount of Cryogenic fluid required to cool metals (litres/kg)

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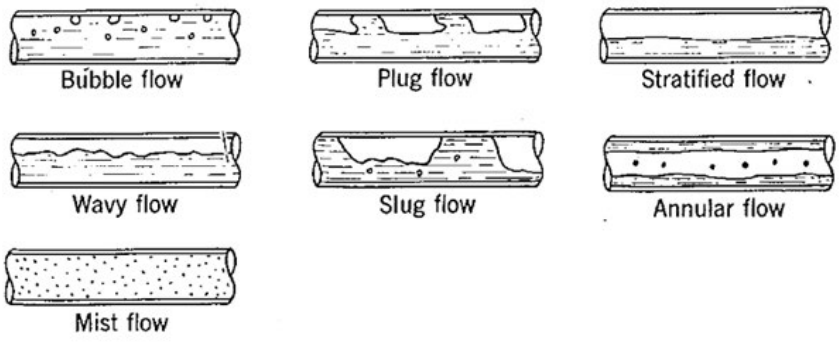
6.9 TWO-PHASE FLOW IN CRYOGENIC FLUID-TRANSFER SYSTEMS:[1]

Two-phase flow is common in cryogenic fluid transfer. Predicting pressure drop in two-phase flow is complicated since the flow patterns are different for horizontal, vertical or inclined flow. And, different flow patterns may exist in a given configuration depending on flow conditions; Also, the 'quality' of fluid may change along the length of pipe due to heat transfer or pressure drop.

Various modes of two-phase flow are shown in the following fig

Fig.6.17 Two-phase flow patterns for horizontal flow.



The **Baker diagram** shown below can be used to predict the flow regime present in a horizontal pipe. It is in SI Units. [13]:

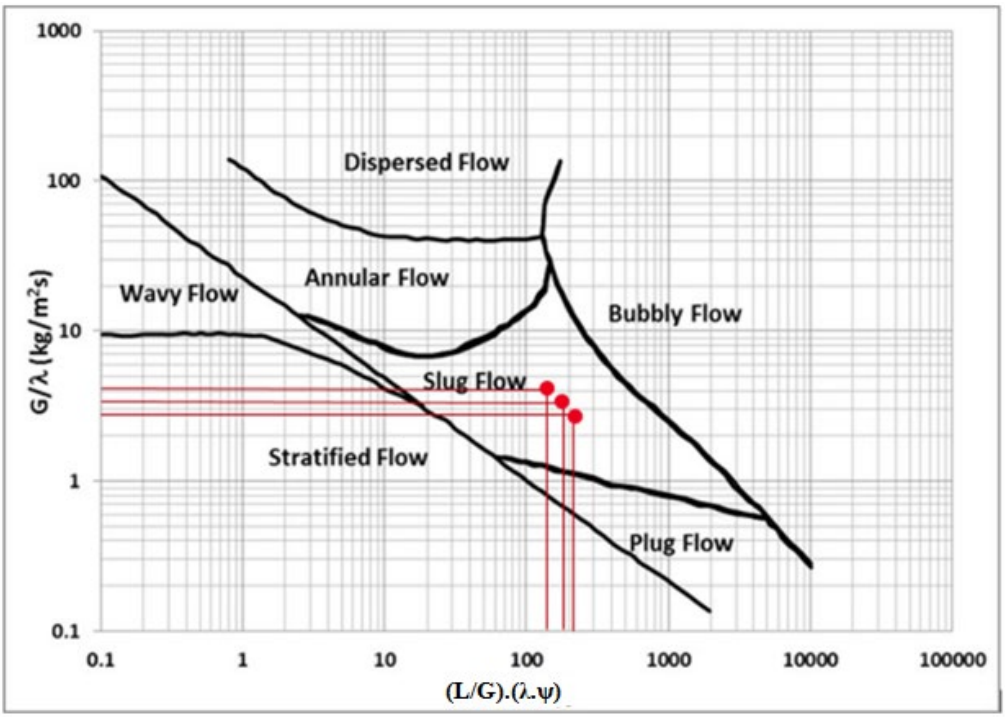


Fig. 6.18 Baker diagram for two-phase flow [13].

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In the above fig:

$\lambda = [(\rho_G/\rho_a)(\rho_L/\rho_w)]^{1/2}$

$\psi = (\sigma_w/\sigma_L)[(\mu_L/\mu_w)(\rho_w/\rho_L)^2]^{1/3}$

L = liquid mass flow rate per unit area, kg/s.m^2

G = vapor mass flow rate per unit area, kg/s.m^2

ρ_G = vapor density kg/m^3

ρ_L = liquid density kg/m^3

ρ_a = density of air = 1.20 kg/m^3 = 0.075 lb$_m$/ft^3

ρ_w = density of water = 998 kg/m^3 = 62.3 lb$_m$/ft^3

σ_L = surface tension N/m

σ_w = water surface tension = 0.073 N/m

μ_L = liquid viscosity Pa-s

μ_w = water viscosity = 0.001 Pa-s = 1 centipoise

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Following Table identifies a few typical flow regimes, with examples, in the Baker diagram shown above:

$G/\lambda\left(kg/\left(m^2 s\right)\right.$	$L\lambda\psi/G$	Flow Regimes from Baker chart, Fig. 1
2	2	Stratified
27	0.2	Wavy
0.2	6000	Plug
4	100	Slug
41	1000	Bubble
109	10	Spray

6.10 PRESSURE DROP IN TWO-PHASE FLOW:[1]

There are generally ***two models*** used to predict pressure drop in two-phase flow: (a) **homogeneous model**, where the liquid and vapor phases are treated as a single phase, with suitably averaged properties for fluids (b) **separated-flow model**, where the two phases are considered to be separated in to two separate streams.

Lockhart and Martinelli (1949) developed a semi-empirical correlation for two-phase flow in horizontal tubes based on separated flow model. This correlation has been used with good accuracy for cryogenic fluids such as hydrogen.

Parameters used for predicting two-phase pressure drop in adiabatic flow according to Lockhart-Martinelli correlation are:

$$\left(\frac{\Delta p}{\Delta L}\right)_{TP} = \phi_L^2 \cdot \left(\frac{\Delta p}{\Delta L}\right)_L$$

$$X^2 = \frac{\left(\frac{\Delta p}{\Delta L}\right)_L}{\left(\frac{\Delta p}{\Delta L}\right)_G} = \frac{C_L \cdot \left(Re_G\right)^m \cdot \rho_G}{C_G \cdot \left(Re_L\right)^n \cdot \rho_L} \cdot \left(\frac{1-x}{x}\right)^2$$

where:

$\left(\frac{\Delta p}{\Delta L}\right)_{TP}$ is the pressure drop per unit length for two-phase flow,

$\left(\frac{\Delta p}{\Delta L}\right)_L$ and $\left(\frac{\Delta p}{\Delta L}\right)_G$ are pressure drops per unit length that would exist if the pipe were completely filled with liquid(L) or vapor(G) alone.

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Reynold's number Re_L and Re_G are calculated from:

$$Re_L = \frac{D \cdot m_L}{A \cdot \mu_L} \quad \text{and,} \quad Re_G = \frac{D \cdot m_G}{A \cdot \mu_G}$$

where:

D = tube inside dia, μ = viscosity, A = cross-sectional area, and m = mass flow rate

Constants C_L, C_G, m and n in the equation for X^2 above are given Table 6.20:

Constant	Laminar Re < 2000	Turbulent 3000 < Re < 50 000	Turbulent Re > 50 000
m (vapor)	1	0.25	0.20
n (liquid)	1	0.25	0.20
C_G (vapor)	64	0.316	0.184
C_L (liquid)	64	0.316	0.184

The four possible combinations are:
vv = laminar liquid, laminar vapor
vt = laminar liquid, turublent vapor
tv = turbulent liquid, laminar vapor
tt = turbulent liquid, turbulent vapor

Table 6.20 Lockhart-Martinelli correlation constants

The Lockhart-Martinelli parameters are related by the following expression:

$$\phi_L = \frac{\left(X^2 + C \cdot X + 1\right)^{\frac{1}{2}}}{X}$$

where: constant C has the values shown in following Table:

Liquid	Vapor	*C*
Laminar	Laminar (*vv*)	5
Turbulent	Laminar (*tv*)	10
Laminar	Turbulent (*vt*)	12
Turbulent	Turbulent (*tt*)	20

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Pressure drop for liquid alone in the pipe is determined from:

$$\left(\frac{\Delta p}{\Delta L}\right)_L = \frac{f \cdot \left(\frac{m_L}{A}\right)^2}{2 \cdot \rho_L \cdot D}$$

The friction factor f is determined from:

$f = \frac{64}{Re}$ .... for Laminar flow, i.e. Re < 2300

$f = 0.316 \cdot Re^{-0.25}$ ....for turbulent flow, 3000 < Re < 50,000

$f = 0.184 \cdot Re^{-0.2}$ ....for turb. flow, Re > 50,000.

$f = \left(1.82 \cdot \log\left(Re\right) - 1.64\right)^{-2}$ ..Petukhov eqn...for turb. flow, 10,000 < Re < 5E06

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Note: To calculate pressure drop in two-phase flow, several calculators are available on line. See, for example:

1. https://www.enggcyclopedia.com/calculators/pressure-drop/pipe-pressure-drop-calculator-phase/
2. http://excelcalculations.blogspot.com
3. https://checalc.com/fluid_flow_two_phase.html
4. https://cheguide.com/two_phase_horiz.html … this site gives some background theory about equations used to determine two-phase pressure drop; also, *one can download for free an excellent EXCEL program to determine the two-phase pressure drop by different methods*.
5. To determine flow regime in two-phase flow: http://www.enground.com/PipingSys/Baker%202%20Phase%20Flow%20Regime.html

Prob.6.26. Liquid nitrogen flows in a 200-mm nominal (8-in. nominal) schedule 5 pipe at a total flow rate of 4.50 kg/s (35 700 lbm/hr) and a fluid **quality of 0.150. The liquid temperature is 85 K (153°R), and the liquid and vapor densities are 771 kg/m3 and 9.376 kg/m3, respectively. The liquid viscosity and surface tension are 0.119 mPa-s and 7.18 mN/m. Determine the two-phase flow regime.[1]**

Solution:

With reference to the Baker diagram shown above, we have to find X and Y coordinates and, then, find out the flow regime.

We have:

X - axis: $\lambda^* (G_L/G_G) * \psi$ Y--axis: G_G/λ

Data:

$D := 213.56 \cdot 10^{-3}$ m ... ID of 200 mm nominal pipe, with OD = 219.1 mm, thickness = 2.77 mm

$A := \frac{\pi \cdot D^2}{4}$ i.e. A = 0.036 m^2 .... area of cross-section

$\rho_a := 1.2$ kg/m^3 ... density of air $\rho_w := 998$ kg/m^3 ... density of water

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$\rho_L := 771$ kg/m^3... density of liq.

$\rho_G := 9.376$ kg/m^3.... density of vap.

$\mu_L := 0.119 \cdot 10^{-3}$ Pa..viscosity of liq.

$\sigma_L := 7.18 \cdot 10^{-3}$ N/m .... surface tension

$x := 0.15$ ....quality

$\sigma_w := 0.073$ N/m .... surface tension of water

$m_{tot} := 4.5$ kg/s ... total flow rate

Therefore: $m_G := 4.5 \cdot 0.15$ kg/s ... vapor flow rate i.e. $m_G = 0.675$ kg/s

$m_L := m_{tot} - m_G$ i.e. $m_L = 3.825$ kg/s.... liquid flow rate

And: $G_L := \frac{m_L}{A}$ i.e. $G_L = 106.783$ kg/s.m^2

$G_G := \frac{m_G}{A}$ i.e. $G_G = 18.844$ kg/s.m^2

Now: $\lambda := \left(\frac{\rho_G}{\rho_a} \cdot \frac{\rho_L}{\rho_w}\right)^{\frac{1}{2}}$ i.e. $\lambda = 2.457$

$$\psi := \left(\frac{\sigma_w}{\sigma_L}\right) \cdot \left[\left(\frac{\mu_L}{\mu_w}\right) \cdot \left(\frac{\rho_w}{\rho_L}\right)^2\right]^{\frac{1}{3}} \qquad \psi = 5.94$$

Then: $\frac{G_L}{G_G} \cdot (\lambda \cdot \psi) = 82.694$ ....X axis in the Baker diagram

and: $\frac{G_G}{\lambda} = 7.67$ ...Y axis in the Baker diagram

Then, from the Baker diagram, we see that: for X = 82.694 and Y = 7.67, the flow regime is: *SLUG FLOW*.

Prob.6.27. Write a Mathcad Function to determine the pressure drop in two-phase flow.

We use the equations given in Barron [1] for the Lockhart-Martinelli correlation.

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$$\text{Lockhart\_Martinelli\_PrDrop2}\left(D, \rho_L, \rho_G, \mu_L, \mu_G, m_L, m_G\right) :=$$

$$A \leftarrow \frac{\pi \cdot D^2}{4}$$

$$G_L \leftarrow \frac{m_L}{A}$$

$$G_G \leftarrow \frac{m_G}{A}$$

$$Re_L \leftarrow \frac{G_L \cdot D}{\mu_L}$$

$$Re_G \leftarrow \frac{G_G \cdot D}{\mu_G}$$

$$m\_vap \leftarrow 1 \quad \text{if} \quad Re_G < 2000$$

$$n\_liq \leftarrow 1 \quad \text{if} \quad Re_L < 2000$$

$$C_G \leftarrow 64 \quad \text{if} \quad Re_G < 2000$$

$$C_L \leftarrow 64 \quad \text{if} \quad Re_L < 2000$$

$$m\_vap \leftarrow 0.25 \quad \text{if} \quad 3000 < Re_G < 50000$$

$$m\_vap \leftarrow 0.2 \quad \text{if} \quad Re_G \geq 50000$$

$$n\_liq \leftarrow 0.25 \quad \text{if} \quad 3000 < Re_L < 50000$$

$$n\_liq \leftarrow 0.2 \quad \text{if} \quad Re_L \geq 50000$$

$$C_G \leftarrow 0.316 \quad \text{if} \quad 3000 < Re_G < 50000$$

$$C_G \leftarrow 0.184 \quad \text{if} \quad Re_G \geq 50000$$

$$C_L \leftarrow 0.316 \quad \text{if} \quad 3000 < Re_L < 50000$$

$$C_L \leftarrow 0.184 \quad \text{if} \quad Re_L \geq 50000$$

$$x \leftarrow \frac{m_G}{m_L + m_G}$$

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$$X_{tt} \leftarrow \left[\frac{C_L \cdot Re_G^{m\_vap} \cdot \rho_G}{C_G \cdot Re_L^{n\_liq} \cdot \rho_L} \cdot \left(\frac{1-x}{x}\right)^2\right]^{\frac{1}{2}}$$

$$C \leftarrow 5 \quad \text{if} \quad (Re_L < 3000) \cdot (Re_G < 3000)$$

$$C \leftarrow 10 \quad \text{if} \quad (Re_L > 3000) \cdot (Re_G < 3000)$$

$$C \leftarrow 12 \quad \text{if} \quad (Re_L < 3000) \cdot (Re_G > 3000)$$

$$C \leftarrow 20 \quad \text{if} \quad (Re_L > 3000) \cdot (Re_G > 3000)$$

$$\phi_L \leftarrow \frac{\left(X_{tt}^2 + C \cdot X_{tt} + 1\right)^{\frac{1}{2}}}{X_{tt}}$$

$$f_L \leftarrow 0.316 \cdot Re_L^{-0.25} \quad \text{if} \quad 50000 > Re_L > 3000$$

$$f_L \leftarrow 0.184 \cdot Re_L^{-0.2} \quad \text{if} \quad Re_L \geq 50000$$

$$f_G \leftarrow 0.184 \cdot Re_G^{-0.2} \quad \text{if} \quad Re_G \geq 50000$$

$$f_G \leftarrow 0.316 \cdot Re_G^{-0.25} \quad \text{if} \quad 50000 > Re_G > 3000$$

$$dPdL_L \leftarrow \frac{f_L}{2} \cdot \frac{G_L^2}{\rho_L \cdot D}$$

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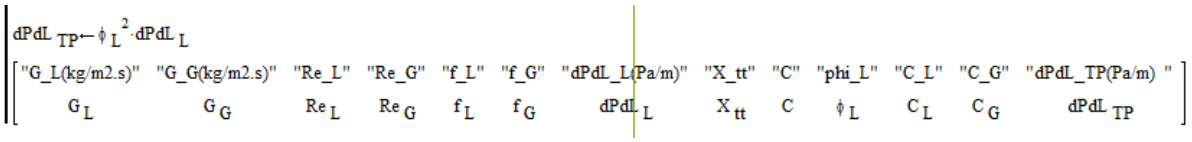
dPdL_TP ← ϕ_L^2 · dPdL_L
"G_L(kg/m2.s)" "G_G(kg/m2.s)" "Re_L" "Re_G" "f_L" "f_G" "dPdL_L(Pa/m)" "X_tt" "C" "phi_L" "C_L" "C_G" "dPdL_TP(Pa/m) "
G_L G_G Re_L Re_G f_L f_G dPdL_L X_tt C ϕ_L C_L C_G dPdL_TP

In the above Function:

Inputs

Subscript L for Liquid, subscript G for Vapor

D = ID of pipe, m; ρ = density, kg/m^3; μ = viscosity, Pa-s; m = mass flow rate, kg/s

Outputs:

G_L = mass flow rate of liquid per unit area, kg/s.m^2

G_G = mass flow rate of vapor per unit area, kg/s.m^2

Re_L = Reynolds Number of Liquid

Re_G = Reynolds Number of vapor

f_L = friction factor for only the liquid flowing full in pipe

f_G = friction factor for only the vapor flowing full in pipe

$dPdL_L$ = Pressure drop per unit length (Pa/m) if only the liquid is flowing

X_{tt} = Lokhart-Martinelli parameter

C = Constant in the expression for the multiplier (φ_L) to $dPdL_L$ to calculate two-phase pressure drop

φ_L = the multiplier to $dPdL_L$ to calculate two-phase pressure drop

C_L = constant to calculate X_{tt}

C_G = constant to calculate X_{tt}

$dPdL_{TP}$ = Two-phase pressure drop per unit length, Pa/m

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Prob.6.28. Determine the frictional pressure drop for adiabatic flow of two-phase hydrogen in a 150-mm nominal (6-in. nominal) schedule 5 pipe for a total mass flow rate of 4.00 kg/s (31,750 lbm/hr) and a fluid **quality of 0.10. The liquid hydrogen is flowing at an average pressure of 354.6 kPa (3.50 atm). The length of the line is 160 m (525 ft).[1]**

Solution:

For hydrogen at 354.6 kPa (saturation temperature = 25.478 K, we find the following fluid properties, from NIST website:

Ref: https://webbook.nist.gov/chemistry/fluid/

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Liquid Phase Data

Data on Saturation Curve

Temperature (K)	Pressure (MPa)	Density (kg/m3)	Volume (m3/kg)	Internal Energy (kJ/kg)	Enthalpy (kJ/kg)	Entropy (J/g*K)	Cv (J/g*K)
25.478	0.35460	63.944	0.015639	55.261	60.807	2.4843	5.9820

Cp (J/g*K)	Sound Spd. (m/s)	Joule-Thomson (K/MPa)	Viscosity (uPa*s)	Therm. Cond. (W/m*K)	Surf. Tension (N/m)	Phase
13.835	944.96	-0.31667	9.1415	0.10373	0.0011292	liquid

Vapor Phase Data

Data on Saturation Curve

Temperature (K)	Pressure (MPa)	Density (kg/m3)	Volume (m3/kg)	Internal Energy (kJ/kg)	Enthalpy (kJ/kg)	Entropy (J/g*K)	Cv (J/g*K)
25.478	0.35460	4.2933	0.23292	380.82	463.41	18.286	6.8055

Cp (J/g*K)	Sound Spd. (m/s)	Joule-Thomson (K/MPa)	Viscosity (uPa*s)	Therm. Cond. (W/m*K)	Phase
15.873	376.29	13.951	1.4188	0.023549	vapor

Data:

$D := 0.1627$ m...ID of 150 mm nominal pipe

$\rho_L := 63.944$ kg/m^3 $\rho_G := 4.2933$ kg/m^3

$\mu_L := 9.1415 \cdot 10^{-6}$ Pa-s $\mu_G := 1.4188 \cdot 10^{-6}$ Pa-s

$m_L := 3.6$ kg/s $m_G := 0.4$ kg/s

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Then, applying the Mathcad Function written above, we get:

Lockhart_Martinelli_PrDrop2$(D, \rho_L, \rho_G, \mu_L, \mu_G, m_L, m_G)$ =

	0	1	2	3	4	5	6
0	"G_L(kg/m2.s)"	"G_G(kg/m2.s)"	"Re_L"	"Re_G"	"f_L"	"f_G"	"dPdL_L(Pa/m)"
1	173.156	19.24	$3.082 \cdot 10^6$	$2.206 \cdot 10^6$	$9.269 \cdot 10^{-3}$	$9.91 \cdot 10^{-3}$	13.357

	7	8	9	10	11	12
	"X_tt"	"C"	"phi_L"	"C_L"	"C_G"	"dPdL_TP(Pa/m) "
	2.255	20	3.172	0.184	0.184	134.429

We note that two-phase pressure drop is: $dPdL_{TP}$ = 134.429 Pa/m

Now, length of pipe is, by data, L = 160 m

Then, total two-phase pressure drop = 134.429 x 160 = 2.151E04 Pa = 21.51 kPa … Ans.

Now, determine the flow regime from the Baker diagram:

We have:

$\rho_a := 1.2$ kg/m^3 ... density of air

$\rho_w := 998$ kg/m^3 ... density of water

$\sigma_w := 0.073$ N/m .... surface tension of water

$\mu_w := 0.001$ Pa-s .... viscosity of water

$\sigma_L := 0.0011292$ N/m .... surface tension of Liquid

Then, we get:

$$\lambda := \left(\frac{\rho_G}{\rho_a} \cdot \frac{\rho_L}{\rho_w}\right)^{\frac{1}{2}} \quad \text{i.e.} \quad \lambda = 0.479$$

$G_G := 19.24$ kg/s.m^2 $G_L := 173.156$ kg/s.m^2

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$$\frac{G_G}{\lambda} = 40.185 \quad \text{kg/s.m^2} \quad \text{.....Y-axis}$$

$$\psi := \left(\frac{\sigma_w}{\sigma_L}\right)\cdot\left[\left(\frac{\mu_L}{\mu_w}\right)\cdot\left(\frac{\rho_w}{\rho_L}\right)^2\right]^{\frac{1}{3}} \quad \text{i.e.} \quad \psi = 84.42$$

Then: $$\left(\lambda\cdot\frac{G_L}{G_G}\cdot\psi\right) = 363.762 \quad \text{....X-axis}$$

Then, refer to Baker diagram. For X = 363.72 and Y = 40.185, we read:

Flow regime is: *Bubble Flow*...Ans.

==========

Prob.6.29 Write a Mathcad program for two-phase flow pressure drop in a pipe; include the effect of roughness too.

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Solution:

We will use the Haaland equation (given below) for friction factor, which includes the effect of pipe roughness.

$$\frac{1}{\sqrt{f}} = -1.8 \log\left[\left(\frac{\varepsilon/D}{3.7}\right)^{1.11} + \frac{6.9}{\mathrm{Re}}\right]$$

Haaland equation is an approximation of the implicit Colebrook-White eqn. (Ref: Wikipedia: https://en.wikipedia.org/wiki/Darcy_friction_factor_formulae#Haaland_equation

This is valid for only turbulent flow i.e. for $Re_D > 4000$.

Some typical roughness values (e) for pipes are given below:

Ref: https://neutrium.net/fluid_flow/absolute-roughness/

Material	Roughness (mm)
Drawn Tubing, Glass, Plastic	0.0015-0.01
Drawn Brass, Copper, Stainless Steel (New)	>0.0015-0.01
Flexible Rubber Tubing - Smooth	0.006-0.07
Flexible Rubber Tubing - Wire Reinforced	0.3-4
Stainless Steel	0.03
Wrought Iron (New)	0.045
Carbon Steel (New)	0.02-0.05
Carbon Steel (Slightly Corroded)	0.05-0.15
Carbon Steel (Moderately Corroded)	0.15-1
Carbon Steel (Badly Corroded)	1-3
Carbon Steel (Cement-lined)	1.5
Asphalted Cast Iron	0.1-1

Material	Roughness (mm)
Cast Iron (new)	0.25
Cast Iron (old, sandblasted)	1
Sheet Metal Ducts (with smooth joints)	0.02-0.1
Galvanized Iron	0.025-0.15
Wood Stave	0.18-0.91
Wood Stave, used	0.25-1
Smooth Cement	0.5
Concrete Very Smooth	0.025-0.2
Concrete Fine (Floated, Brushed)	0.2-0.8
Concrete Rough, Form Marks	0.8-3
Riveted Steel	0.91-9.1
Water Mains with Tuberculations	1.2
Brickwork, Mature Foul Sewers	3

Moody's Chart: gives Re vs f for various values of 'relative roughness, (= e/D)':

Ref: https://en.m.wikipedia.org/wiki/Moody_chart

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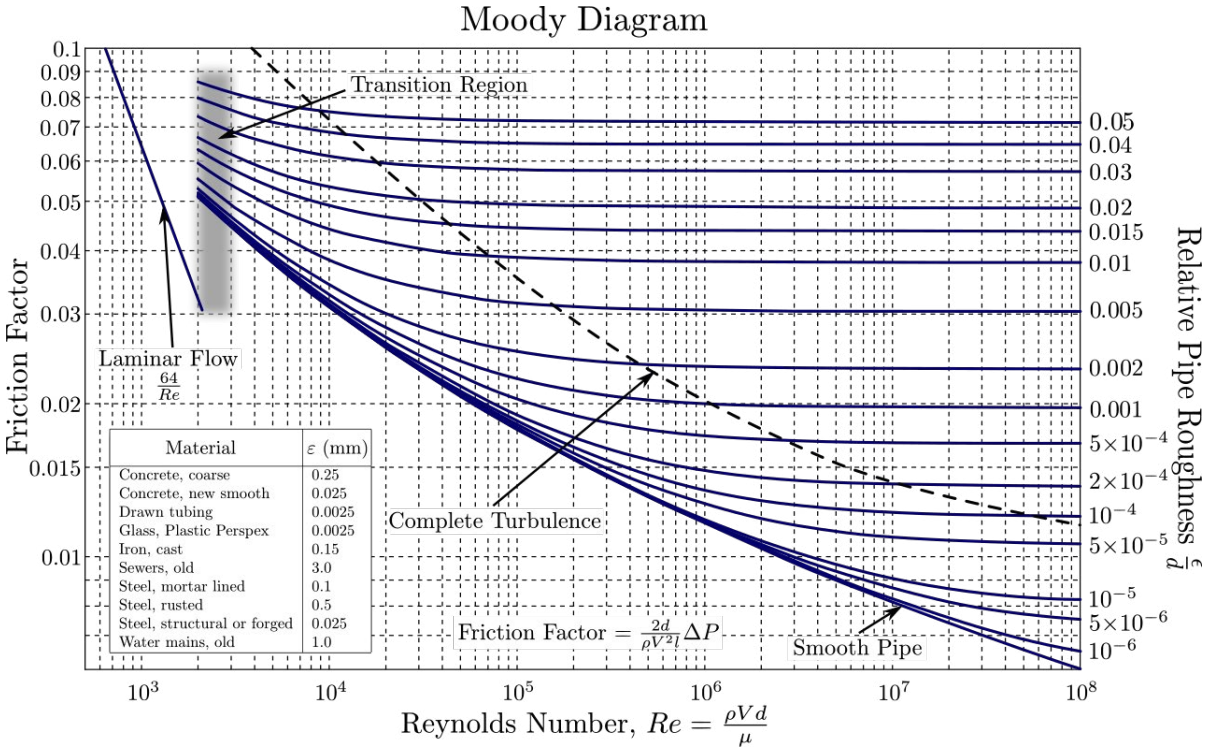

With this background, let us now write the **Mathcad Function** to determine the pressure drop in two-phase flow:

Inputs:

ε = roughness of pipe, m; D = ID of pipe, m; ρ_L = density of liquid, kg/m3;

ρ_G = density of vapor, kg/m3; μ_L = viscosity of liquid, Pa-s; μ_G = viscosity of vapor, Pa-s;

m_L = mass flow rate of liquid, kg/s; m_G = mass flow rate of vapor, kg/s

Outputs:

A = area of cross-section of pipe, m^2; G_L = mass flow rate of liq. / unit area, kg/s.m2;

G_G = mass flow rate of vap. / unit area, kg/s.m2; Re_L = Reynold's number for liquid flow;

Re_G = Reynold's number for vapor flow; ε = roughness of pipe, m;

f_L = friction factor for liquid flow; f_G = friction factor for vapor flow;

$dPdL_L$ = pressure drop for liquid, per unit length, Pa/m;

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$dPdL_L$ = pressure drop for vapor, per unit length, Pa/m;

X_{tt} = Lockhart-Martinelli parameter

φ_L = multiplication factor for liquid flow to get pressure drop in two-phase flow;

φ_G = multiplication factor for vapor flow to get pressure drop in two-phase flow;

$dPdL_{tot1}$ = Two-phase pressure drop based on liquid flow, Pa/m

$dPdL_{tot1}$ = Two-phase pressure drop based on vapor flow, Pa/m

$$\text{Lockhart\_Martinelli\_PrDrop}\left(\varepsilon, D, \rho_L, \rho_G, \mu_L, \mu_G, m_L, m_G\right) :=$$

$$A \leftarrow \frac{\pi \cdot D^2}{4}$$

$$G_L \leftarrow \frac{m_L}{A}$$

$$G_G \leftarrow \frac{m_G}{A}$$

$$Re_L \leftarrow \frac{G_L \cdot D}{\mu_L}$$

$$Re_G \leftarrow \frac{G_G \cdot D}{\mu_G}$$

$$\text{return} \begin{bmatrix} \text{"Note:"} & \text{"ReL / ReG less than 4000 !!"} \\ \text{" Re\_L"} & \text{"Re\_G"} \\ Re_L & Re_G \end{bmatrix} \text{ if } \left[\left\langle Re_L < 4000 \right\rangle - \left\langle Re_G < 4000 \right\rangle\right]$$

$$f_L \leftarrow \left[\frac{1}{\left[-1.8 \cdot \log\left[\left(\frac{\varepsilon}{D \cdot 3.7}\right)^{1.11} + \frac{6.9}{Re_L}\right]\right]}\right]^2 \text{ if } Re_L \geq 4000$$

$$f_G \leftarrow \left[\frac{1}{\left[-1.8 \cdot \log\left[\left(\frac{\varepsilon}{D \cdot 3.7}\right)^{1.11} + \frac{6.9}{Re_G}\right]\right]}\right]^2 \text{ if } Re_G \geq 4000$$

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$$dPdL_L \leftarrow \frac{f_L}{2} \cdot \frac{G_L^2}{\rho_L \cdot D}$$

$$dPdL_G \leftarrow \frac{f_G}{2} \cdot \frac{G_G^2}{\rho_G \cdot D}$$

$$X_{tt} \leftarrow \left(\frac{dPdL_L}{dPdL_G}\right)^{\frac{1}{2}}$$

$$\phi_G \leftarrow \left(1 + 18 \cdot X_{tt} + X_{tt}^2\right)^{\frac{1}{2}}$$

$$dPdL_{tot1} \leftarrow \phi_G^2 \cdot dPdL_G$$

$$\phi_L \leftarrow \left(1 + \frac{18}{X_{tt}} + \frac{1}{X_{tt}^2}\right)^{\frac{1}{2}}$$

$$dPdL_{tot2} \leftarrow \phi_L^2 \cdot dPdL_L$$

"G_L(kg/m2.s)"	"G_G(kg/m2.s)"	"Re_L"	"Re_G"	"epsilon(m)"	"fL"	"fG"	"dPdL_L(Pa/m)"	"dPdL_G(Pa/m)"	"Xtt"	"phi_L"	"phi_G"	"dPdLtot1(Pa/m)"	"dPdLtot2(Pa/m)"
G_L	G_G	Re_L	Re_G	ε	f_L	f_G	$dPdL_L$	$dPdL_G$	X_{tt}	ϕ_L	ϕ_G	$dPdL_{tot1}$	$dPdL_{tot2}$

Now, let us solve Prob.6.28 with this Mathcad Function:

Prob.6.30. Determine the frictional pressure drop for adiabatic flow of two-phase hydrogen in a 150-mm nominal (6-in. nominal) schedule 5 pipe for a total mass flow rate of 4.00 kg/s (31,750 lbm/hr) and a fluid quality of 0.10. The liquid hydrogen is flowing at an average pressure of 354.6 kPa (3.50 atm). The length of the line is 160 m (525 ft).[1]

We have:

$D := 0.1627$ m $\quad \varepsilon := 0.0$

$\rho_L := 63.944$ kg/m^3 $\quad \rho_G := 4.2933$ kg/m^3

$\mu_L := 9.1415 \cdot 10^{-6}$ Pa-s $\quad \mu_G := 1.4188 \cdot 10^{-6}$ Pa-s

$m_L := 3.6$ kg/s $\quad m_G := 0.4$ kg/s

Note that we have put value for roughness of pipe ε = 0, since it is a smooth pipe.

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Applying the Mathcad Function:

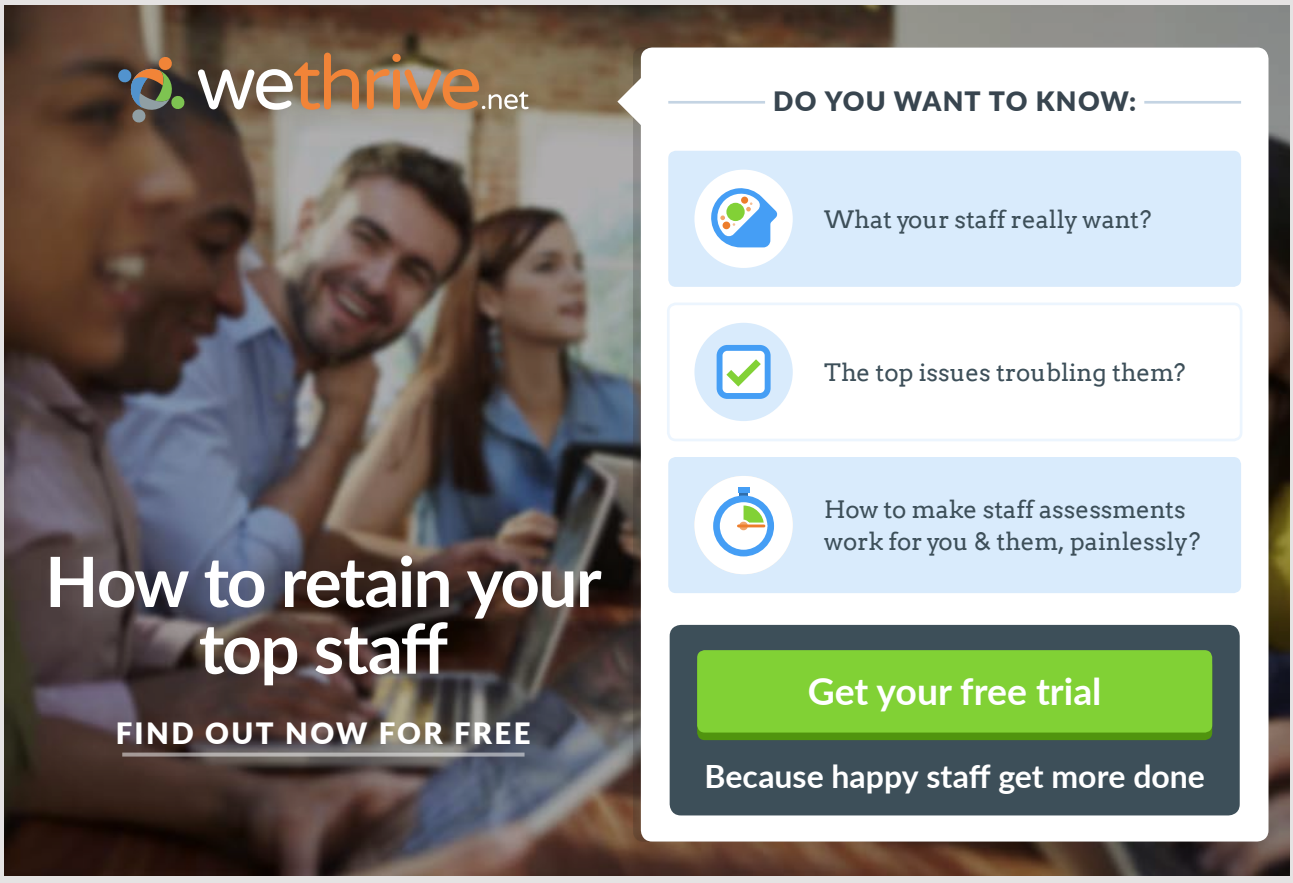
$$\text{Lockhart\_Martinelli\_PrDrop}\left(\varepsilon, D, \rho_L, \rho_G, \mu_L, \mu_G, m_L, m_G\right) =$$

	0	1	2	3	4	5	6	7
0	"G_L(kg/m2.s)"	"G_G(kg/m2.s)"	"Re_L"	"Re_G"	"epsilon(m)"	"fL"	"fG"	"dPdL_L(Pa/m)"
1	173.156	19.24	$3.082 \cdot 10^{6}$	$2.206 \cdot 10^{6}$	0	$9.669 \cdot 10^{-3}$	0.01	13.932

8	9	10	11	12	13
"dPdL_G(Pa/m)"	"Xtt"	"phi_L"	"phi_G"	"dPdLtot1(Pa/m)"	"dPdLtot2(Pa/m)"
2.699	2.272	3.019	6.86	127.003	127.003

Note that now, the two-phase pressure drop is 127.003 Pa/m, whereas earlier, in Prob.6.18, it was 134.29 Pa/m. .. (not much different!)

(b) If the roughness of pipe is 0.001 m, what will be new pressure drop?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So, we have:

$D := 0.1627$ m $\quad \varepsilon := 0.001$ m.... roughness of pipe

$\rho_L := 63.944$ kg/m^3 $\quad \rho_G := 4.2933$ kg/m^3

$\mu_L := 9.1415 \cdot 10^{-6}$ Pa-s $\quad \mu_G := 1.4188 \cdot 10^{-6}$ Pa-s

$m_L := 3.6$ kg/s $\quad m_G := 0.4$ kg/s

And,

$\text{Lockhart\_Martinelli\_PrDrop}\left(\varepsilon, D, \rho_L, \rho_G, \mu_L, \mu_G, m_L, m_G\right) =$

	0	1	2	3	4	5	6	7
0	"G_L(kg/m2.s)"	"G_G(kg/m2.s)"	"Re_L"	"Re_G"	"epsilon(m)"	"fL"	"fG"	"dPdL_L(Pa/m)"
1	173.156	19.24	$3.082 \cdot 10^6$	$2.206 \cdot 10^6$	$1 \cdot 10^{-3}$	0.032	0.032	46.756

8	9	10	11	12	13
"dPdL_G(Pa/m)"	"Xtt"	"phi_L"	"phi_G"	"dPdLtot1(Pa/m)"	"dPdLtot2(Pa/m)"
8.6	2.332	2.984	6.958	416.299	416.299

We get: two-phase pressure drop = 416.299 Pa/m … Ans.

==========

Prob.6.31. Determine the frictional pressure drop for adiabatic flow of Liquid nitrogen in a 200-mm nominal (8-in. nominal) schedule 5 pipe for a total flow rate of 4.50 kg/s (35 700 lbm/hr) and a fluid quality of 0.150. The liquid temperature is 85 K (153°R), and the length of pipe line is 300 m (984 ft).[1]

Solution:

Properties of Nitrogen at sat. temp of 85 K:

Ref: https://webbook.nist.gov/chemistry/fluid/

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Liquid Phase Data

Data on Saturation Curve

Temperature (K)	Pressure (MPa)	Density (kg/m3)	Volume (m3/kg)	Internal Energy (kJ/kg)	Enthalpy (kJ/kg)	Entropy (J/g*K)	Cv (J/g*K)
85.000	0.22886	770.13	0.0012985	-106.46	-106.16	3.0277	1.0429

Cp (J/g*K)	Sound Spd. (m/s)	Joule-Thomson (K/MPa)	Viscosity (uPa*s)	Therm. Cond. (W/m*K)	Surf. Tension (N/m)	Phase
2.0906	772.44	-0.28135	121.11	0.12993	0.0071627	liquid

Vapor Phase Data

Data on Saturation Curve

Temperature (K)	Pressure (MPa)	Density (kg/m3)	Volume (m3/kg)	Internal Energy (kJ/kg)	Enthalpy (kJ/kg)	Entropy (J/g*K)	Cv (J/g*K)
85.000	0.22886	9.8241	0.10179	59.056	82.352	5.2454	0.79094

Cp (J/g*K)	Sound Spd. (m/s)	Joule-Thomson (K/MPa)	Viscosity (uPa*s)	Therm. Cond. (W/m*K)	Phase
1.1957	179.68	22.923	6.0405	0.0083518	vapor

Now, let us write all data:

$D := 213.56 \cdot 10^{-3}$ m ... ID of 200 mm nominal pipe, with OD = 219.1 mm, thickness = 2.77 mm

$A := \frac{\pi \cdot D^2}{4}$ i.e. A = 0.036 m^2 .... area of cross-section

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$\rho_L := 770.13$ kg/m^3 $\rho_G := 9.8241$ kg/m^3

$\mu_L := 121.11 \cdot 10^{-6}$ Pa-s $\mu_G := 6.0405 \cdot 10^{-6}$ Pa-s

$m_{tot} := 4.5$ kg/s ... total flow rate $x := 0.15$ ...quality of fluid flowing

$m_G := m_{tot} \cdot x$ kg/s i.e. $m_G = 0.675$ kg/s

$m_L := m_{tot} - m_G$ i.e. $m_L = 3.825$ kg/s

$\varepsilon := 0$ ...roughness for smooth pipe = 0

Now, apply the simple Mathcad Function for two-phase pressure drop, using Haaland eqn, written above:

$\text{Lockhart\_Martinelli\_PrDrop}(\varepsilon, D, \rho_L, \rho_G, \mu_L, \mu_G, m_L, m_G) =$

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	0	1	2	3	4	5	6	7
0	"G_L(kg/m2.s)"	"G_G(kg/m2.s)"	"Re_L"	"Re_G"	"epsilon(m)"	"fL"	"fG"	"dPdL_L(Pa/m)"
1	106.783	18.844	$1.883 \cdot 10^5$	$6.662 \cdot 10^5$	0	0.016	0.012	0.544

8	9	10	11	12	13
"dPdL_G(Pa/m)"	"Xtt"	"phi_L"	"phi_G"	"dPdLtot1(Pa/m)"	"dPdLtot2(Pa/m)"
1.051	0.719	5.288	3.803	15.203	15.203

Thus, we see that two-phase pressure drop is 15.203 Pa/m. i.e.

$\Delta P_{tot} := 15.203$ Pa/m

Then, for a 300 m long pipe, the pressure drop is:

$\Delta P_{300m} := \Delta P_{tot} \cdot 300$ Pa .....i.e. $\Delta P_{300m} = 4.561 \cdot 10^3$ Pa = 45.61 kPa ....Ans.

==========

Prob.6.32. Two-phase liquid oxygen flows in a 200-mm nominal (8-in. nominal) schedule 10 pipe at a total mass flow rate of 10 kg/s (79,400 lb“,/hr) and a quality of 0.200. The length of the line is 275 m (902 ft), and the fluid **temperature is 110 K. The vapor density is 19.40 kg/m3, and the liquid density is 1035.4 kg/m3. Determine the two-phase frictional pressure drop.**[1]

Solution:

We will take properties of Oxygen at sat. temp of 110 K from NIST website:

Ref: https://webbook.nist.gov/chemistry/fluid/

Liquid Phase Data

Data on Saturation Curve

Temperature (K)	Pressure (MPa)	Density (kg/m3)	Volume (m3/kg)	Internal Energy (kJ/kg)	Enthalpy (kJ/kg)	Entropy (J/g*K)	Cv (J/g*K)
110.00	0.54340	1035.5	0.00096574	-99.165	-98.641	3.2855	0.86576

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Cp (J/g*K)	Sound Spd. (m/s)	Joule-Thomson (K/MPa)	Viscosity (uPa*s)	Therm. Cond. (W/m*K)	Surf. Tension (N/m)	Phase
1.8068	734.77	-0.19639	122.72	0.12249	0.0083869	liquid

Vapor Phase Data

Data on Saturation Curve

Temperature (K)	Pressure (MPa)	Density (kg/m3)	Volume (m3/kg)	Internal Energy (kJ/kg)	Enthalpy (kJ/kg)	Entropy (J/g*K)	Cv (J/g*K)
110.00	0.54340	21.281	0.046990	65.520	91.054	5.0100	0.69881

Cp (J/g*K)	Sound Spd. (m/s)	Joule-Thomson (K/MPa)	Viscosity (uPa*s)	Therm. Cond. (W/m*K)	Phase
1.1014	188.14	17.250	9.0060	0.010795	vapor

Now, let us write all data:

$D := 211.58 \cdot 10^{-3}$ m ... ID of 200 mm nominal pipe, with OD = 219.1 mm, thickness = 3.76 mm for sch 10 pipe

$A := \frac{\pi \cdot D^2}{4}$ i.e. A = 0.035 m^2 .... area of cross-section

Properties of sat. liquid and vapor Oxygen from NIST website:

$\rho_L := 1035.5$ kg/m^3 $\rho_G := 21.281$ kg/m^3

$\mu_L := 122.72 \cdot 10^{-6}$ Pa-s $\mu_G := 9.006 \cdot 10^{-6}$ Pa-s

$m_{tot} := 10$ kg/s ... total flow rate $x := 0.2$ ...quality of fluid flowing

$m_G := m_{tot} \cdot x$ kg/s i.e. $m_G = 2$ kg/s

$m_L := m_{tot} - m_G$ i.e. $m_L = 8$ kg/s

$\varepsilon := 0$ ...roughness for smooth pipe = 0

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Now, apply the simple Mathcad Function for two-phase pressure drop, using Haaland eqn:

We get:

$\text{Lockhart\_Martinelli\_PrDrop}\left(\varepsilon, D, \rho_L, \rho_G, \mu_L, \mu_G, m_L, m_G\right) =$

	0	1	2	3	4	5	6
0	"G_L(kg/m2.s)"	"G_G(kg/m2.s)"	"Re_L"	"Re_G"	"epsilon(m)"	"fL"	"fG"
1	227.536	56.884	$3.923 \cdot 10^5$	$1.336 \cdot 10^6$	0	0.014	0.011

7	8	9	10	11	12	13
"dPdL_L(Pa/m)"	"dPdL_G(Pa/m)"	"Xtt"	"phi_L"	"phi_G"	"dPdLtot1(Pa/m)"	"dPdLtot2(Pa/m)"
1.613	3.967	0.638	5.629	3.589	51.116	51.116

Thus, we see that two-phase pressure drop is 51.116 Pa/m.

Therefore, for a length of 275 m, the pressure drop is:

$\Delta P_{275m} := 51.116 \cdot 275$ Pa

i.e. $\Delta P_{275m} = 1.406 \cdot 10^4$ Pa **= 14.06 kPa .... Ans.**

==========

6.11 COOL DOWN OF CRYOGENIC LINES:[1]

Line is initially at ambient temperature and as the liquid enters the line, it quickly boils, and the vapor continues down the line. Various types of two phase flow exist in the line. Severe pressure and flow oscillations occur. Liquid front moves down the line, and when the entire line is filled with liquid, it is said that the cool down is completed.

In the analysis of cool down process, the space within the transfer line is the 'thermodynamic system'. See fig. below:

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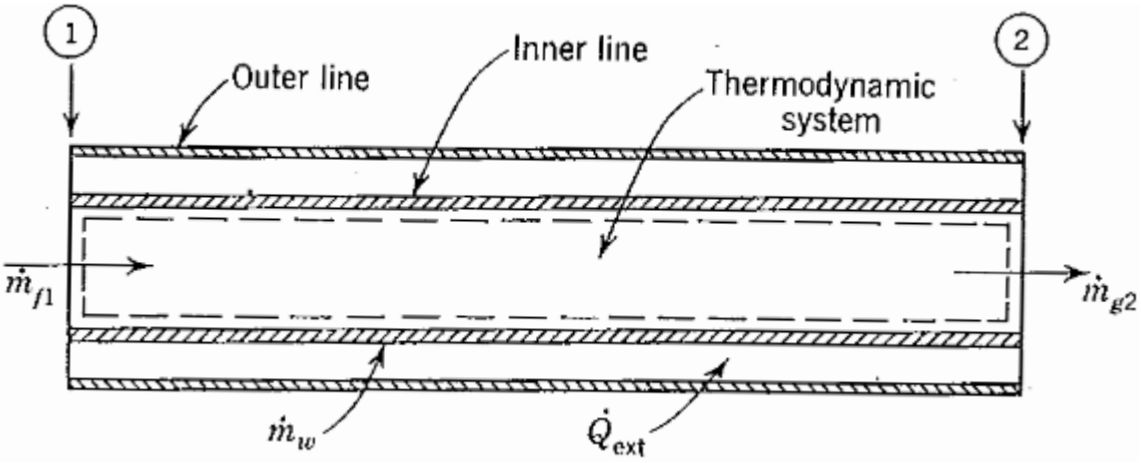

Applying the I Law, we have:

$$Q=\left(m_{ss}\cdot u_{ss}-m_o\cdot u_o\right)+\int_0^{t_{ss}}\left(m_{g2}\cdot h_{g2}\right)dt-\int_0^{t_{ss}}\left(m_{f1}\cdot h_{f1}\right)dt$$

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where: Q="total heat transferred to system"

m="mass contained within the system"

u="int. energy of fluid within the system"

m_{g2}="mass flow rate of fluid from system"

h_{g2}="enthalpy of fluid leaving the system"

t_{ss}="cool down time"

m_{f1}="mass flow rate of liquid into the line"

h_{f1}="enthalpy of liquid entering line"

Total heat transferred to the system, Q:

$$Q=\int_0^{t_{ss}} Q\,dt=\int_0^{t_{ss}} Q_{ext}\,dt+\Sigma\cdot\int_{T_{ss}}^{T_o} m_w\cdot c_w\,dT_w$$

where:

Q_{ext}="heat tr rate from ambient"

m_w="mass of solid portions of line (inner line, valve bodies etc"

c_w="sp. heat of solid portions of line"

The first integral above may be approximated by an average heat transfer rate equal to one-half of final steady state heat transfer rate, Q_{ss} and the second integral may be written in terms of the mean specific heats of the solid material of the line, cw_avg:

i.e.

$$Q=\frac{1}{2}\cdot Q_{ss}\cdot t_{ss}+\Sigma\cdot m_w\cdot c_{w\_avg}\cdot\left(T_o-T_{ss}\right)$$

Applying the conservation of mass to the system:

$$m_{ss}-m_0=\int_0^{t_{ss}} m_{f1}\,dt-\int_0^{t_{ss}} m_{g2}\,dt=\left(m_{f1\_avg}-m_{g2\_avg}\right)\cdot t_{ss}$$

where: m_{f1_avg} and m_{g2_avg} are mean flow rates of liq into the system and vapor out of the system.

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Also, define a mean enthalpy at the exit, hg2_avg by:

$$m_{g2} \cdot h_{g2\_avg} \cdot t_{ss} = \int_0^{t_{ss}} m_{g2} \cdot h_{g2}\, dt$$

With these definitions, the I Law may be written as:

$$Q = \left[m_{ss} \cdot \left(u_{ss} - h_{f1} \right) - m_o \cdot \left(u_o - h_{f1} \right) \right] + m_{g2\_avg} \cdot \left(h_{g2\_avg} - h_{f1} \right) \cdot t_{ss}$$

The initial and final mass terms may be written in terms of internal volume of the line, V and the fluid densities:

$$m_o = V \cdot \rho_{go}$$

$$m_{ss} = V \cdot \rho_{ss}$$

where:

$$\frac{1}{\rho_{ss}} = \frac{\left(1 - x_{avg}\right)}{\rho_f} + \frac{x_{avg}}{\rho_g}$$

where: ρ_f = "sat. liq. density"

ρ_g = "sat. vap. density"

ρ_{go} = "density of gas initially in the line"

$x_{avg} = \frac{x_1 + x_2}{2}$ = mean quality of fluid under steady state conditions

Making these substitutions and solving for cool down time, we get:

$$t_{ss} = \frac{\Sigma\left[m_w \cdot c_{w\_avg} \cdot \left(T_o - T_{ss} \right) \right] - V \cdot \left[\rho_{ss} \cdot \left(u_{ss} - h_{f1} \right) - \rho_{go} \cdot \left(u_o - h_{f1} \right) \right]}{m_{g2} \cdot \left(h_{g2\_avg} - h_{f1} \right) - 0.5 \cdot Q_{ss}}$$

Several additional approximations are made:

Mean enthalpy of vapor leaving the line, h_{g2}_avg, is determined from the exit gas pressure and the mean temperature, T_{g2}_avg:

$$T_{g2\_avg} = \frac{T_{ss} + T_o}{2}$$ ..mean temp of exit gas

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During cool down, flow is usually 'choked' at the exit, i.e. mean vapor velocity at exit is sonic velocity, V2, given by:

$$V2\_sonic = C_d \cdot \left(\gamma \cdot R \cdot T_{g2\_avg}\right) \quad \text{m/s , Cd = discharge coeff}$$

where: , C_d = discharge coeff at the termination of line

γ = sp. heat ratio

R = sp. gas constant

Total mass of liquid required to cool down:

This is given by:

$$m_f = V \cdot \left(\rho_{ss} - \rho_{go}\right) + m_{g2\_avg} \cdot t_{ss}$$

The first term on the RHS gives the mass of liquid required to fill the line; the second term is the mass of liquid evaporated in cooling down the line and in absorbing the heat transfer from the ambient during the cool down. Generally, the second term is much smaller than the first term.

So, neglecting this term and solving for the **specific liquid requirement** from the equation for t_{ss}, we get:

$$\frac{m_f}{M_w} = \frac{V \cdot \left(\rho_{ss} - \rho_{go}\right)}{M_w} + \frac{Q_{ss} \cdot t_{ss}}{2 \cdot M_w \cdot \left(h_{g2\_avg} - h_{f1}\right)} + \frac{C_{w\_avg} \cdot \left(T_o - T_{ss}\right)}{h_{g2\_avg} - h_{f1}}$$

where:

$M_w = \Sigma m_w$ = total mass of transfer line, valves, joints etc.

$C_w = \frac{\left(\Sigma m_w \cdot c_{w\_avg}\right)}{M_w}$ = mean thermal capacity of the transfer line system

The last term in the above equation represents the mass of liquid required to cool down the transfer system per unit mass of the system.

Note: In connection with the liquid requirements, recollect that in Ref.[12], volume of cryogenic liquid required to cool Aluminum, Stainless Steel and copper (in litres/kg) are

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given (i) when using the latent heat of vaporization only, and (ii) when using the enthalpy of evaporated gas also. **See Table below:**

The volume of liquid helium required to cool a mass of metal from one temperature to another using the full enthalpy of the gas or using the latent heat alone was reported by J B Jacobs, Advances in Cryogenic Engineering, Volume 8, 1963, p. 529, as follows.

Cryogen		**^{4}He**	**^{4}He**	**N_2**
Initial temperature of metal		300 K	77 K	300 K
Final temperature of metal		4.2 K	4.2 K	77 K
Using the latent heat	Al	66.6	3.20	1.01
of vaporisation only *	St. Steel	33.3	1.43	0.53
	Cu	31.1	2.16	0.46
Using the enthalpy	Al	1.61	0.22	0.64
of the gas **	St. Steel	0.79	0.11	0.33
	Cu	0.79	0.15	0.29

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Prob.6.33. Determine the cool down time and liquid requirement for a 100 mm nominal, sch. 5 inner line transfer line. Length of line is 75 m, and the line contains 5 bayonet joints (total mass = 45 kg) and one valve (mass = 12 kg). The line is initially at 300 K and is filled with nitrogen gas at 101.3 kPa and 300 K. LN2 at 354.6 kPa is used to cool the line to a final temperature of 89.8 K. At steady state operation, exit quality is 0.006 and heat in-leak from ambient is 6 W/m. The entire line and valve are constructed of SS 304 having a mean specific heat of 0.375 kJ/kg.K. The valve at the end of the line has a free cross-sectional area of 0.0018 m^2 and a discharge coeff. of 0.6. [1]

Solution:

Data:

100 mm nominal, sch. 5 pipe has dimensions: OD = 114.3 mm, t = 2.1 mm; Therefore, ID= 110.1 mm

$D_o := 0.1143$ m $D_i := 0.1101$ m $t := 0.0021$ m

$A := 0.0018$ m^2..... valve cross-sectional area at the end of the line... by data

$L := 75$ m ... length of line $\rho := 7920$ kg/m^3 ... for SS 304

$c_{w\_avg} := 0.375$ kJ/kg.K .... avg sp. heat of metals

$T_o := 300$ K .... initial line temp, K

$T_{ss} := 89.8$ K ... final steady state temp of line, K

$C_d := 0.6$ ...discharge coeff for valve at exit

$\gamma := 1.4$. ratio of sp. heats for Nitrogen $R := 296.8$ kJ/kg.K ... gas constant for N2

$Q_{ss} := 6 \cdot L \cdot 10^{-3}$ i.e. $Q_{ss} = 0.45$ kW ... ext heat leak for L=75 m at the rate of 6 W/m...by data

Mass of inner line:

$$m_w := \frac{\pi \cdot \left(D_o^2 - D_i^2\right)}{4} \cdot L \cdot \rho \qquad \text{i.e.} \qquad m_w = 439.6919 \text{ kg}$$

Total mass to be cooled:

$$M_w := m_w + 45 + 12 \qquad \text{i.e.} \qquad M_w = 496.6919 \text{ kg}$$

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Therefore, total energy to be removed from metal:

$$Q_{steel} := M_w \cdot c_{w\_avg} \cdot \left(T_o - T_{ss}\right)$$

i.e. $Q_{steel} = 3.91517 \cdot 10^4$ kJ

Total volume of inner line, V:

$$V := \frac{\pi}{4} \cdot (0.1101)^2 \cdot L \quad m^3$$

i.e. $V = 0.714$ m^3

Mean gas temp at exit, Tg2_avg:

$$T_{g2\_avg} := \frac{T_o + T_{ss}}{2} \quad K$$

i.e. $T_{g2\_avg} = 194.9$ K

Therefore, mean exit velocity (sonic):

$$V2\_sonic := C_d \cdot \left(\gamma \cdot R \cdot T_{g2\_avg}\right)^{\frac{1}{2}} \quad m/s$$

i.e. $V2\_sonic = 170.747$ m/s

At the exit pressure of 101.3 kPa and exit temp of 194.9 K, mean exit density, rg2 is:

Isobaric Data for P = 0.10130 MPa

Temperature (K)	Pressure (MPa)	Density (kg/m3)	Volume (m3/kg)	Internal Energy (kJ/kg)	Enthalpy (kJ/kg)	Entropy (J/g*K)	Cv (J/g*K)	Cp (J/g*K)	Sound Spd. (m/s)	Joule-Thomson (K/MPa)	Viscosity (uPa*s)	Therm. Cond. (W/m*K)	Phase
194.90	0.10130	1.7554	0.56968	143.97	201.67	6.3923	0.74309	1.0438	284.37	4.8999	12.647	0.018259	vapor

i.e. $\rho_{g2} := 1.7554$ kg/m^3 ... from NIST website

Then, mean gas flow rate at the exit is:

$$m_{g2} := \rho_{g2} \cdot A \cdot V2\_sonic \quad \text{i.e.} \quad m_{g2} = 0.53951 \text{ kg/s}$$

Assuming that the liquid enters the line as sat. liquid, and the exit quality is 0.006, we have the mean quality = 0.003

i.e. $x_{avg} := 0.003$

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And the final fluid density is given by:

$$\frac{1}{\rho_{ss}} = \frac{(1 - x_{avg})}{\rho_f} + \frac{x_{avg}}{\rho_g}$$

where: $\rho_f := 746$ kg/m^3 .... sat. liq density at inlet, i.e. at 356.4 kPa .... from NIST website

$\rho_g := 14.846$ kg/m^3 .... sat. vap. density at inlet, i.e. at 356.4 kPa.... from NIST website

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See below:

At 354.6 kPa:

Liquid Phase Data

Data on Saturation Curve

Temperature (K)	Pressure (MPa)	Density (kg/m3)	Volume (m3/kg)	Internal Energy (kJ/kg)	Enthalpy (kJ/kg)	Entropy (J/g*K)	Cv (J/g*K)	Cp (J/g*K)	Sound Spd. (m/s)	Joule-Thomson (K/MPa)	Viscosity (uPa*s)	Therm. Cond. (W/m*K)	Surf. Tension (N/m)	Phase
89.809	0.35460	746.01	0.0013405	-96.404	-95.928	3.1428	1.0205	2.1385	721.08	-0.22808	103.16	0.12027	0.0061263	liquid

Vapor Phase Data

Data on Saturation Curve

Temperature (K)	Pressure (MPa)	Density (kg/m3)	Volume (m3/kg)	Internal Energy (kJ/kg)	Enthalpy (kJ/kg)	Entropy (J/g*K)	Cv (J/g*K)	Cp (J/g*K)	Sound Spd. (m/s)	Joule-Thomson (K/MPa)	Viscosity (uPa*s)	Therm. Cond. (W/m*K)	Phase
89.809	0.35460	14.846	0.067359	60.997	84.883	5.1561	0.80713	1.2624	181.72	20.896	6.4428	0.0089931	vapor

Now, enthalpy of sat. liq at 354.6 kPa, 89.8 K:

$h_{fl} := -95.928$ kJ/kg .... from NIST website

Also: $u_f := -96.404$ kJ/kg .... from NIST website

$u_g := 60.997$ kJ/kg .... from NIST website

Therefore, int. energy in steady state, u_ss is:

$$u_{ss} := u_f + x_{avg} \cdot \left(u_g - u_f\right)$$

i.e. $u_{ss} = -95.9318$ kJ/kg

At the initial condition of 101.3 kPa and 300 K, we get, from NIST website, the gas density ρo and the int. energy, u_go as:

$\rho_{go} := 1.1379$ kg/m^3

$u_o := 222.17$ kJ/kg

See below: Density of N2 at 101.3 kPa, 300K:

Isobaric Data for P = 0.10130 MPa

Temperature (K)	Pressure (MPa)	Density (kg/m3)	Volume (m3/kg)	Internal Energy (kJ/kg)	Enthalpy (kJ/kg)	Entropy (J/g*K)	Cv (J/g*K)	Cp (J/g*K)	Sound Spd. (m/s)	Joule-Thomson (K/MPa)	Viscosity (uPa*s)	Therm. Cond. (W/m*K)	Phase
300.00	0.10130	1.1379	0.87882	222.17	311.19	6.8418	0.74317	1.0414	353.16	2.1196	17.897	0.025858	vapor

Mean enthalpy of gas at exit condition of 101.3 kPa, 194.9K, from NIST website:

$h_{g2\_avg} := 201.67$ kJ/kg

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See below: Mean enthalpy of gas at exit condition of 0.1013 MPa, 194.9K:

Isobaric Data for P = 0.10130 MPa

Temperature (K)	Pressure (MPa)	Density (kg/m3)	Volume (m3/kg)	Internal Energy (kJ/kg)	Enthalpy (kJ/kg)	Entropy (J/g*K)	Cv (J/g*K)	Cp (J/g*K)	Sound Spd. (m/s)	Joule-Thomson (K/MPa)	Viscosity (uPa*s)	Therm. Cond. (W/m*K)	Phase
194.90	0.10130	1.7554	0.56968	143.97	201.67	6.3923	0.74309	1.0438	284.37	4.8999	12.647	0.018259	vapor

Now, let us find out the cool down time:

$$t_{ss} = \frac{\Sigma\left[m_w \cdot c_{w\_avg} \cdot \left(T_o - T_{ss}\right)\right] - V \cdot \left[\rho_{ss} \cdot \left(u_{ss} - h_{f1}\right) - \rho_{go} \cdot \left(u_o - h_{f1}\right)\right]}{m_{g2} \cdot \left(h_{g2\_avg} - h_{f1}\right) - 0.5 \cdot Q_{ss}}$$

i.e. $$t_{ss} := \frac{Q_{steel} - V \cdot \left[\rho_{ss} \cdot \left(u_{ss} - h_{f1}\right) - \rho_{go} \cdot \left(u_o - h_{f1}\right)\right]}{m_{g2} \cdot \left(h_{g2\_avg} - h_{f1}\right) - 0.5 \cdot Q_{ss}}$$

i.e. $t_{ss} = 245.8133$ seconds = **4.1 min. ... Ans.**

Specific cool down requirements:

1. Specific mass required to fill the line with liquid:

$$\left(\frac{m_f}{M_w}\right)_l = \frac{V \cdot \left(\rho_{ss} - \rho_{go}\right)}{M_w} = 0.944 \quad \text{kg of liq/kg of SS304}$$

2. Specific mass evaporated due to ambient heat transfer:

$$\left(\frac{m_f}{M_w}\right)_q = \frac{Q_{ss} \cdot t_{ss}}{2 \cdot M_w \cdot \left(h_{g2\_avg} - h_{f1}\right)} = 3.742 \cdot 10^{-4} \quad \text{kg of liq/kg of SS304}$$

3. Specific mass required for cool down of line is:

$$\frac{m_f}{M_w} = \frac{c_{w\_avg} \cdot \left(T_o - T_{ss}\right)}{h_{g2\_avg} - h_{f1}} = 0.26487 \quad \text{kg of liq/kg of SS304}$$

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Thus, the liquid requirements are:

To fill the line:

$m_f = 0.944 \cdot M_w = 468.877$ **kg.. to fill the line ... Ans.**

To take care of ext. heat in-leak:

$m_f = 0.0003742 \cdot M_w = 0.186$ **kg.. to take care of heat in-leak....Ans.**

To cool down the line:

$m_f = 0.26487 \cdot M_w = 131.559$ **kg.. to cool down the line....Ans.**

==========

Note: From the Table given above this problem statement, we see that amount of LN2 required (with cooling with the enthalpy of gas also included) is 0.33 litres/kg of SS 304. So, for 497 kg, we get : amount of LN2 required for cool down = 497 * 0.33 = 164.01 litres = 164.01 * 0.8 kg = 131.21 kg. This matches very well with the figure of 131.559 kg obtained with detailed calculations above.

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Prob.6.34. Determine the time required to cool down a 30 m (98.40-ft) section of 100-mm nominal (4-in. nominal) stainless steel transfer line used to transfer liquid hydrogen. The mean specific heat of the stainless steel is 315 J /kg-K, and the inner line has a total mass of 180 kg (397 lb). The inner line is initially at 295 K (531°R) and is cooled down to a final temperature of 20.5 K (36.9°R). The average heat transfer rate from ambient during the cooling process is (Q_{ss}/2) = 70 W (239 Btu/hr). The average gas enthalpy at the exit of the line is 2270 kJ/kg, and the average mass flow rate of the gas from the line during the cool-down process is 20 g/s. The internal energy of the gas initially within the line is 2938 kJ/kg, and the density of the gas initially within the line is 0.0832 kg/m3. The density of the liquid finally within the line is 70.51 kg/m3, and the final internal energy of the fluid in the line is 27l.7 kJ/kg. The enthalpy of the liquid entering the line is 273.2 kJ/kg. Also, determine the mass of liquid required to fill the line and to cool down the line.[1]

Solution:

Here, most of the variables required to calculate cool down time and liquid requirements, are given.

So, let us write a general Mathcad Function to calculate the cool down time etc. when LN2 or LH2 are used:

Note: Fluid 1 ---> "Nitrogen" Fluid 2 ---> "Hydrogen"

Gas constant R and avg. sp. heat of SS304 are different when LN2 or LH2 is used. This is taken care in the first two blocks of the program below.

Rest of the inputs and outputs are already defined in the problem above.

$$\text{Cool\_down\_time\_cryoline}\left(\text{Fluid}, M_w, L, D, T_o, T_{ss}, Q_{ext}, h_{g2\_avg}, m_{g2\_avg}, u_o, \rho_{go}, \rho_{ss}, u_{ss}, h_{fl}\right) :=$$

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```
if Fluid="Nitrogen"
    R ← 296.8
    c_w_avg ← 0.375
if Fluid="Hydrogen"
    R ← 4124
    c_w_avg ← 0.315
Q_steel ← M_w · c_w_avg · (T_o − T_ss)
V ← (π · D^2 / 4) · L
t_ss ← (Q_steel − V · [ρ_ss · (u_ss − h_f1) − ρ_go · (u_o − h_f1)]) / (m_g2_avg · (h_g2_avg − h_f1) − 0.5 · Q_ext)
m_f_to_fill_the_line ← V · (ρ_ss − ρ_go)
m_f_towards_heatinleak ← (Q_ext · t_ss) / (2 · (h_g2_avg − h_f1))
m_f_to_cooldown_line ← M_w · (c_w_avg · (T_o − T_ss)) / (h_g2_avg − h_f1)
```

"Fluid"	"Q_steel (kJ)"	"cool down time, t_ss(sec)"	"mf_to_fill_line (kg)"	"mf_towards_heat_inleak (kg)"	"mf_to_cool_line (kg)"
Fluid	Q_{steel}	t_{ss}	$m_{f\_to\_fill\_the\_line}$	$m_{f\_towards\_heatinleak}$	$m_{f\_to\_cooldown\_line}$

Now, write the data given in the above problem:

Fluid := "Hydrogen" $M_w := 180$ kg $L := 30$ m $D := 0.1101$ m $T_o := 295$ K $T_{ss} := 20.5$ K

$Q_{ext} := 0.14$ kW $h_{g2\_avg} := 2270$ kJ/kg $m_{g2\_avg} := 0.02$ kg/s $u_o := 2938$ kJ/kg $\rho_{go} := 0.0832$ kg/m^3

$\rho_{ss} := 70.51$ kg/m^3 $u_{ss} := 271.7$ kJ/kg $h_{f1} := 273.2$ kJ/kg

And, applying the above Function, we get:

$$Cool\_down\_time\_cryoline(Fluid, M_w, L, D, T_o, T_{ss}, Q_{ext}, h_{g2\_avg}, m_{g2\_avg}, u_o, \rho_{go}, \rho_{ss}, u_{ss}, h_{f1}) =$$

"Fluid"	"Q_steel (kJ)"	"cool down time, t_ss(sec)"	"mf_to_fill_line (kg)"	"mf_towards_heat_inleak (kg)"	"mf_to_cool_line (kg)"
"Hydrogen"	$1.5564 \cdot 10^4$	392.7578	20.1152	0.0138	7.7945

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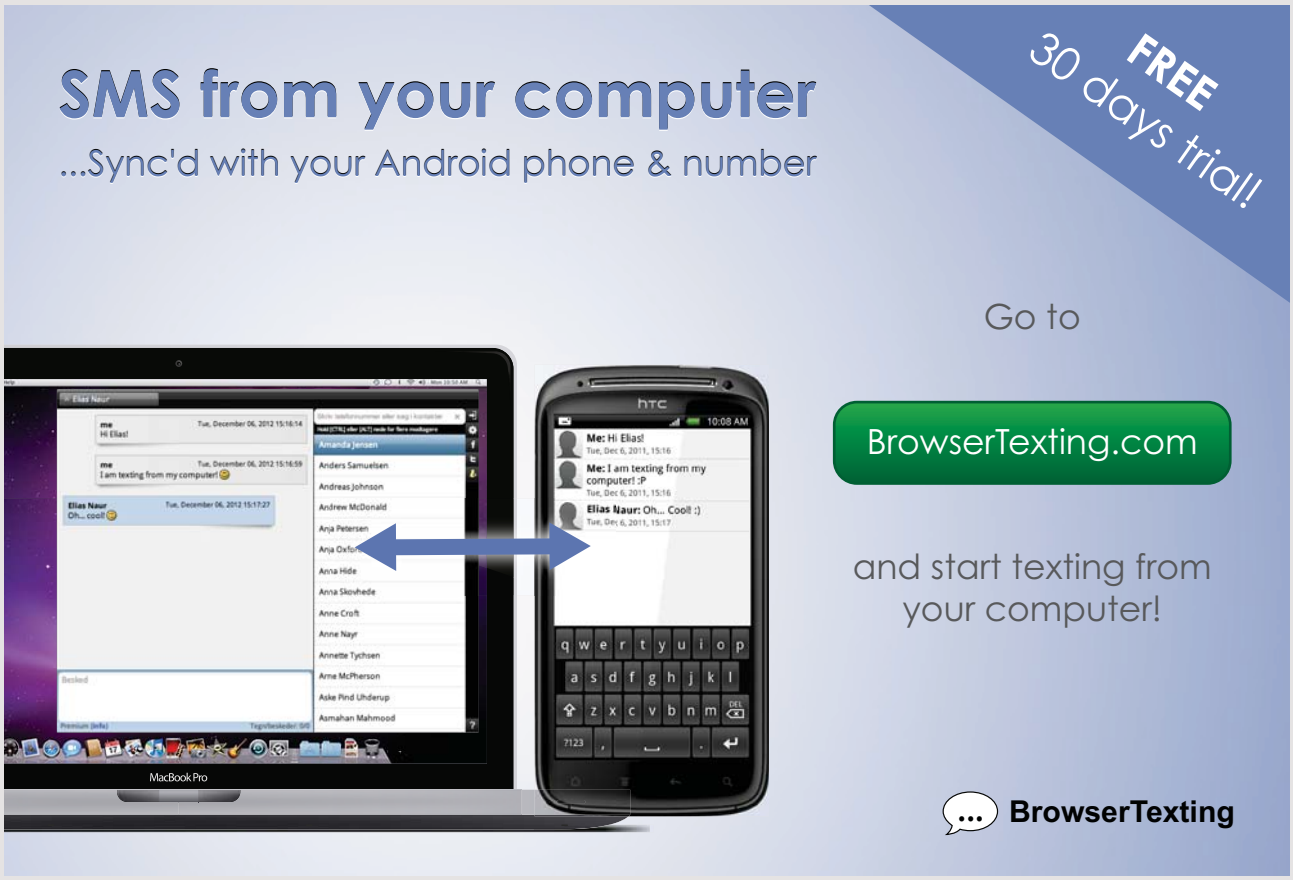
Thus, the results are:

1. **Cool down time = t_ss = 392.76 sec = 6.55 min…. Ans.**
2. **Liquid requirements:**

To fill the line: 20.115 kg

Towards heat in-leak: 0.0138 kg

To cool the line: 7.7945 kg



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Prob.6.35. Solve Prob.6.33 using the above Mathcad Function:

Let us write the required parameters from the solution of Prob.6.33:

$Fluid := "Nitrogen"$ $M_w := 496.7\ kg$ $L := 75\ m$ $D := 0.1101\ m$ $T_o := 300\ K$ $T_{ss} := 89.8\ K$

$Q_{ext} := 0.45\ kW$ $h_{g2\_avg} := 201.67\ kJ/kg$ $m_{g2\_avg} := 0.53951\ kg/s$ $u_o := 222.17\ kJ/kg$ $\rho_{go} := 1.1379\ kg/m^3$

$\rho_{ss} := 657.5237\ kg/m^3$ $u_{ss} := -95.9318\ kJ/kg$ $h_{fl} := -95.928\ kJ/kg$

Now, using the Mathcad Function, we get:

$$Cool\_down\_time\_cryoline\left(Fluid, M_w, L, D, T_o, T_{ss}, Q_{ext}, h_{g2\_avg}, m_{g2\_avg}, u_o, \rho_{go}, \rho_{ss}, u_{ss}, h_{fl}\right) =$$

"Fluid"	"Q_steel (kJ)"	"cool down time, t_ss(sec)"	"mf_to_fill_line (kg)"	"mf_towards_heat_inleak (kg)"	"mf_to_cool_line (kg)"
"Nitrogen"	$3.9152 \cdot 10^4$	245.8187	468.6892	0.1859	131.5613

Note that the results are matching.

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